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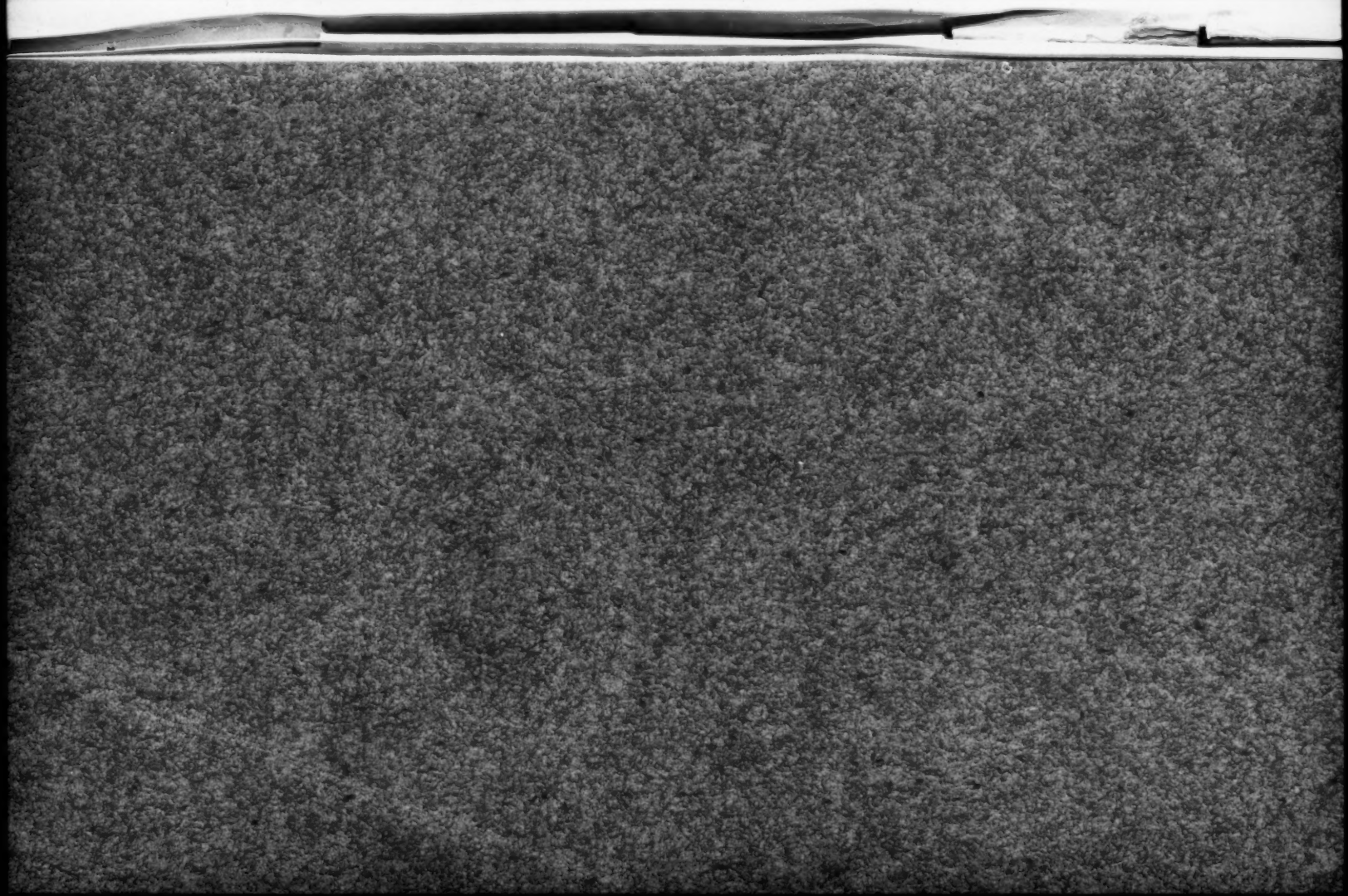

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The Society as a body is not responsible for the statements of facts or opinions advanced in papers or discussions. C55.

TRAINING WORKMEN IN HABITS OF INDUSTRY AND COÖPERATION

By H. L. Gantt, Pawtucket, R. I.

Member of the Society

The widespread interest in the training of workmen which has been so marked for several years is due to the evident need for better methods of training than those now generally in vogue.

2 The one point in which these methods as a class seem to be lacking is that they do not lay enough stress on the fact that workmen must have industry as well as knowledge and skill.

3 *Habits of industry* are far more valuable than any kind of knowledge or skill, for with such habits as a basis, the problem of acquiring knowledge and skill is much simplified. Without industry knowledge and skill are of little value, and sometimes a great detriment.

4 If workmen are systematically trained in habits of industry, it has been found possible, not only to train many of them to be efficient in whatever capacity they are needed, but to develop an effective system of coöperation between workmen and foremen.

This is not a theory, but the record of a fact.

5 It is too much to hope, however, that the methods about to be described will be adopted extensively in the near future, for the great majority of managers, whose success is based mainly on their personal ability, will hesitate before adopting what seems to them the slower and less forceful policy of studying problems and training workmen; but should they do so they will have absolutely no desire to return to their former methods.

6 The general policy of the past has been to drive, but the era of force must give way to that of knowledge, and the policy of the future will be to teach and to lead, to the advantage of all concerned. The vision of workmen in general eager to coöperate in carrying out the

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results of scientific investigations must be dismissed as a dream of the millennium, but results so far accomplished indicate that nothing will do more to bring about that millennium than training workmen in habits of industry and coöperation. A study of the principles on which such training has been successfully established will convince the most skeptical that if they are carried out the results must follow. An outline of these principles has already been submitted to the Society in a paper entitled A Bonus System of Rewarding Labor.[1]

7 Under this system each man has his work assigned to him in the form of a task to be done by a prescribed method with definite appliances and to be completed within a certain time. The task is based on a detailed investigation by a trained expert of the best methods of doing the work; and the task setter, or his assistant, acts as an instructor to teach the workmen to do the work in the manner and time specified. If the work is done within the time allowed by the expert, and is up to the standard for quality, the workman receives extra compensation in addition to his day's pay. If it is not done in the time set, or is not up to the standard for quality, the workman receives his day's pay only.

8 This system, in connection with the other work of Mr. F. W. Taylor, so greatly increased the output and reduced the cost of the work in the large machine shop of the Bethlehem Steel Company that for the past seven years the writer has given a large portion of his time to the development of its possibilities. The results have far exceeded his expectations.

9 In his closing remarks on the above paper, the writer emphasized the value of the system as a means of training workmen, and the late Dr. Robert H. Thurston in his discussion of it was so optimistic as to the results it would produce on "workmen and foremen and employer alike" that the writer felt that his enthusiasm over a new and promising method had carried him, perhaps, a little too far. Results have fully justified Dr. Thurston's predictions, however, for today the method has been developed as a practical system of education and training for all, from the highest to the lowest. The fact, so repeatedly emphasized by Mr. Taylor, *that tasks should be set only as the result of a scientific investigation*, has proven of an educational

[1] A Bonus System of Rewarding Labor, December 1901, a system of task work with a bonus which had recently been introduced by the writer into the large machine shop of the Bethlehem Steel Company, as a part of the system of management, being introduced into their works by Mr. F. W. Taylor.

value hardly to be over-estimated, for the scientific investigation of a process that has been developed without the assistance of science almost always reveals inconsistencies which it is possible to eliminate, thus perfecting the process and at the same time reducing its cost.

10 It is this scientific investigation that points to improvement in methods and educates owners and managers; but the average workman is interested only in his daily wage and has no special desire to learn improved methods. The results of our investigations are of little practical value, therefore, unless we can first teach our workmen how to use them, and then can induce them to do as they are taught.

PRACTICAL APPLICATION

11 For this purpose *an instructor, a task and a bonus* have been found most useful. People as a rule prefer to work at the speed and in the manner to which they have been accustomed, but are usually willing to work at any reasonable speed and in any reasonable manner, if sufficient inducement is offered for so doing, and if they are so trained as to be able to earn the reward. In carrying out this plan we try to find men who are already skilled and able to perform the task set. It frequently happens, however, that the number of such men is insufficient and it takes time to train the unskilled to a proper degree of efficiency; but with a bonus as an incentive, and a proper instructor, a very fair proportion of the unskilled finally succeed in performing a task that was at first entirely beyond them.

12 Unskilled workmen, who under these conditions have become skilled in one kind of work, readily learn another, and soon begin to realize that they can, in a measure at least, make up for their loss in not having learned a trade.

13 As they become more skilled, they form better habits of work, lose less time and become more reliable. Their health improves, and the improvement in their general appearance is very marked. This improvement in health seems to be due to a more regular and active life, combined with a greater interest in their work, for it is a well known fact that *work in which we are interested and which holds our attention without any effort on our part, tires us much less than that we have to force ourselves to do.* The task with a reward for its accomplishment produces this interest and holds the attention, with the invariable results of more work, better work and better satisfied workers.

14 The Task and Bonus method of training not only furnishes the workmen with the required knowledge, but by offering an induce-

ment to utilize that knowledge properly, trains him in proper habits of work.

HABITS OF WORK

15 In all work both *quantity* and *quality* must be considered, and our task method demands a maximum quantity, all of which must be up to the standard for quality. Workmen trained under this method acquire the habit of doing a large amount of work well, and disprove the oft-repeated fallacy that good work must be done slowly. As a matter of fact, our quickest workers almost always do the best work when following instructions. We set great store by the habit of working quickly, for no matter how much skill a workman may have, he will not attain the best success without quickness.

16 *Habits of work* in a mechanic are comparable with *habits of thought* in an engineer, and our industrial schools should make proper habits of work the basis on which to build their training in manual dexterity. The engineering school does not make engineers, but tries to furnish its graduates with an equipment that will enable them to utilize readily and rapidly their own experience and that of others. In the same manner, industrial training schools should equip their graduates with habits of industry that will make them as mechanics capable and willing to do a large amount of good work.

17 As the writer sees it, one of the most valuable assets that the graduate of a technical college or an industrial school can have *is the habit of doing promptly and to the best of his ability the work set before him.* With this habit and reasonable intelligence he can make good progress. This habit is one of the first results of the Task and Bonus system, for it is a noticeable fact that task workers form habits of industry which they maintain even when on day's work with no bonus in sight.

18 In all schemes for technical or industrial education or training that the writer has seen, emphasis has been laid on the importance of knowing how. The writer wishes to add that ability and willingness to do are of at least equal importance. Many skilled workmen make their skill an excuse for slow work, and unless when they are taught how to do they are also taught to do efficiently, they never attain the success that should be theirs.

19 Under our task system the workman is *taught how* and *trained to do* at the same time. *Knowing* and *doing* are thus closely associated in his mind, and it is our experience that the habit of doing efficiently

what is laid out for him becomes so fixed that he performs without hesitation tasks at which a man not trained to follow instructions would absolutely fail. This is exactly what should be expected and means nothing more than that in our industrial army the workman who has gained confidence in his superior follows his orders without hesitation, just as the private soldier follows the orders of his officer even though he does not see where they lead.

20 This is not a fanciful comparison, for I have known more than one case in which a workman expressed his doubts as to the possibility of doing a task, and on getting the reply that the task was all right, said, "If you say it can be done, I will do it."

21 Workers who have been unable to perform their tasks in the time set have frequently asked to have an instructor stand by them with a stop-watch to time the detail operations and show them just wherein they failed, with the result that they soon learned to earn their bonus regularly.

22 The first essential for a workman to become successful under our task system is to obey *orders*, and having acquired this habit he soon finds out that a skilled investigator can learn more about doing a piece of work than he knows "off hand." Having satisfied himself on this point, he goes to work at the tasks set him with the determination to earn his bonus, with the result, if he has the natural ability, that he soon becomes a rapid and skillful workman.

23 Learning to obey orders is often the hardest part of the workman's task, for a large percentage of men seem so constituted as to be apparently unable to do as they are told. As a rule, however, this is a feature of a certain stage of their development only, which under proper conditions they overcome at a later date. For instance, many very capable men who were impatient of restraint when they should have learned a trade, find themselves at the age of twenty-five or less in the class of unskilled workmen, although their ability would have enabled them to do well at almost any trade. It is this class of men, when they have come to realize the difference between a skilled workman and one not skilled, that furnishes us with many of our best task workers. Such men often see in our *instructor, task, bonus* a chance to redeem some of their earlier errors, and by learning thoroughly how to do, and doing one thing after another, in the best way that can be devised, get in a short time a training that does much to make up for the previous neglect of their opportunities.

BOSSES AS SERVANTS AND TEACHERS

24 In a shop operated on this system, where each workman has his task, one man whom we term a *gang boss* usually tends a group of workmen, supplying them with work and appliances and removing the work when finished. Such a man is paid a bonus for each workman who earns a bonus, and an extra bonus if all of his group earn their bonuses. The result is that so long as the workmen perform their tasks, though nominally their boss, he is really their servant, and becomes the boss only when a workman fails to perform his task. The loss of money to the gang boss in case a workman fails to earn his bonus is such that he constantly has his eye on the poor workman and helps him all he can. If, however, he finds that the workman is incapable of being taught, he uses his influence to have a better man put in his place.

25 In starting a shop on task work, an instructor who is capable of teaching each workman how to perform his task must be constantly on hand, and must as a rule teach one workman at a time. This instructor may be the man who has investigated the work and set the task, or he may simply be an instructor capable of following out the work of such an investigator, but he must be readily available as long as any of the workmen need his services, for we make it a rule not to ask a man to do anything in a certain manner and time unless we are prepared to show him how to do it as we specify.

TASK SETTING

26 A task must always be set for performing a definite operation in a specific manner, a minimum time being set for its accomplishment. As compensation, the workman is paid for the time set plus a percentage (usually 20 to 50) of that time, provided the work is done in the time allowed or less. If the time taken is more than the time allowed, the workman gets his day's pay only. The fact that in setting the task the manner of performing the operation is specified enables us to set another task for the same operation if we develop a better or quicker method.

27 If after having performed his task a workman wishes to suggest a quicker or better method for doing the same work, he is given an opportunity if possible to demonstrate his method. If the suggested method really proves to be quicker or better, it is adopted as the standard, and the workman is given a suitable reward. *No*

workman, however, is allowed to make suggestions until he has first done the work in the manner and time specified.

28 It is the duty of the investigator to develop methods and set tasks, and unless the methods developed by him are pretty generally a great deal better than those suggested by the workmen, he is not retained in the position. Working at tasks is pretty good training for task setting, and the writer has gotten more than one task setter from the ranks of task doers.

29 Inasmuch as, after a satisfactory method has been established, a large proportion of the work of the task setter is the study of the time in which operations can be performed, he is popularly known as the *Time Study* man. This term has lead to a misconception of his duties and has caused many honest people to claim that they were putting in our methods when they have put a stop watch in the hands of a bright clerk and told him to find out how quickly the best men were doing certain work. Unquestionably they have in many cases been able to set more accurate piece rates by this method than they had been able to set by the older methods, but they are still far from our ideal, in which the best expert available investigates the work, standardizes the appliances and methods and sets a task that involves utilizing them to their very best efficiency. While the stop watch is often used to establish a method, it is used to determine the time needed to do the work only when the standard methods and appliances are used efficiently. Stop watch observations on work done inefficiently or with ill-adapted appliances, or by poor methods, is absurd and serves only to bring into disrepute all work in which the stop watch is used. Moreover, such use of the stop watch justly excites the contempt and opposition of the workman.

30 To make real and permanent progress, the expert must be able to standardize appliances and methods and write up such instructions as will enable an intelligent workman to follow them. Such standards become permanent, and if the workman is paid a proper bonus for doing the work in the manner and time set, he not only helps maintain the standards, but soon begins to exert his influence to help the progress of standardization.

STANDARDIZATION

31 All work, and all knowledge, for that matter, may be divided into two classes: *Expert* and *Standard*. Expert knowledge may be described as that which has not been reduced to writing in such a

manner as to be generally available, or exists only in the minds of a few. By analogy, expert work is work, the methods of doing which either are known only to a few or have not been so clearly described as to enable a man familiar with that class of work to understand them.

32 On the other hand, standard methods are those that are generally used, or have been so clearly described and proved that a man familiar with that class of work can understand them and safely employ them.

33 The largest problem of our expert is to standardize expert methods and knowledge. When a method has been standardized, a task may be set, and by means of *an instructor and a bonus* a method of maintaining that standard permanently may be established. With increasing efficiency on the part of the workman the standard always has a tendency to become higher.

34 We have here the workman and the foreman using their efforts to maintain standards, for both fail to obtain a bonus if the standard is not maintained. This is so different from the case in which the standard is maintained only by the man in authority with a club that there can be no comparison.

35 From workmen trained under these methods, we get a good supply of instructors and foremen, and occasionally an investigator. From our investigators, who standardize our methods and appliances, we get our superintendents, and our system of management thus becomes self-perpetuating.

36 The superintendent who believes that the sovereign cure for all troubles is to go into the shop and raise a row, has no place under our methods, for when the task and bonus has been established, errors are far more frequent in the office than in the shop, and the man who is given to bluffing soon finds that his methods produce no effect on men that are following written instructions.

OBSTACLES

37 Among the obstacles to the introduction of this system is the fact that it forces everybody to do his duty. Many people in authority want a system that will force everybody else to do his duty, but will allow them to do as they please.

38 The Task and Bonus system when carried out properly is no respecter of persons, and the man who wishes to force the workman to do his task properly must see that the task is properly set and that

proper means are available for doing it. It is not only the workman's privilege, but his duty, to report whatever interferes with his earning his bonus, and the loss of bonus soon educates him to perform this duty no matter how disagreeable it is at first. We investigate every loss of bonus, and place the blame where it belongs. Sometimes we find it belongs pretty high up, for the man who has neglected his duty under one system of management is pretty apt to neglect it at first under another. He must either learn to perform his duty or yield his place, for the pressure from those who lose money by his neglect or incompetence is continuous and insistent.

39 This becomes evident as soon as the task and bonus gets fairly started and the effect is that opposition to its extension develops on the part of all who are not sure of making good under it, or whose expert knowledge is such that they fear it will all soon be standardized. The opposition of such people however is bound to give way sooner or later, for the really capable man and the true expert welcome these methods as soon as they understand them.

HELPS

40 The fact that the task and the bonus enable us to utilize our knowledge and maintain our standards, and that the setting of tasks after a scientific investigation must necessarily not only increase our knowledge but standardize it, brings to our assistance the clearest thinkers and hardest workers in any organization. Our greatest help, however, comes from the workmen themselves. The most intelligent soon realize that we really mean to help them advance themselves, and the ambitious ones welcome the aid of our instructor to remove obstacles that have been in their way for perhaps years. As soon as one such man has earned his bonus for several days, there is usually another man ready to try the task, and unless there is a great lack of confidence on the part of the men in the management, the sentiment rapidly grows in favor of our task work.

DAY WORK AND PIECE WORK

41 As used by the writer, the Task and Bonus System of pay is really a combination of the best features of both day and piece-work. The workman is assured his day rate while being taught to perform his task, and as the bonus for its accomplishment is a percentage of the time allowed, the compensation when the task has been performed is a fixed quantity, and is thus really the equivalent of a piece rate.

Our method of payment then is piece-work for the skilled, and day-work for the unskilled, it being remembered that if there is only work enough for a few, it will always be given to the skilled. This acts as a powerful stimulus to the unskilled, and all who have any ambition try to get into the bonus class. This cannot be too clearly borne in mind, *for we have here all the advantages of day-work combined with those of piece-work without the disadvantage of either*, for the day-worker who has no ambition to become a bonus-worker usually of his own accord seeks work elsewhere, and our working force soon becomes composed of Bonus-Workers, and day-workers who are trying to become Bonus-Workers.

COÖPERATION

42 When 25 per cent of the workers in a plant are Bonus-Workers, they, with those who are striving to get into their class, control the sentiment, and a strong spirit of coöperation develops. This spirit of coöperation in living up to the standards set by the experts, which is the only way a bonus can be earned, benefits the employer by the production of

More work.
Better work.
Cheaper work.

It benefits the workmen by giving them

Better wages.
Increased skill.
Better habits of work.
More pleasure and pride in their work.

43 Not the least important of these results is the fact that the workmen take more pride in their work, for this of itself insures good work. As an instance of this pride, the writer has known girls working under the task system to form a society, admission to which was confined to those that could earn bonus on their work; the workers themselves thus putting a premium on industry and efficiency.

44 The fact that we get better work as well as quicker work seems inconceivable to some. The reasons are:

a Careful inspection, for no bonus is paid unless the work is up to the standard.

b Work done by a prescribed method, and always in the same way.

c Attention needed to do high-speed work, which keeps the mind of the worker on what he is doing and soon results in exceptional skill.

45 The development of skilled workmen by this method is sure and rapid, and wherever the method has been properly established, *the problem of securing satisfactory help has been solved.* During the past few years while there has been so much talk about the "growing inefficiency of labor," the writer has repeatedly proved the value of this method in increasing its efficiency, and the fact that the system works automatically, when once thoroughly established, puts the *possibility of training their own workmen within the reach of all manufacturers.*

TRAINING HELP A FUNCTION OF MANAGEMENT

46 Any system of management that did not make provision for obtaining proper materials to work with would be thought very lax. The day is not far distant when any management that does not make provision for training the workmen it needs will not be regarded as much better, for it is by this means only that a system of management can be made permanent.

47 To be satisfied to draw skilled workmen from the surplus of other plants means as a rule that second rate men only are wanted, and indicates a lack of appreciation of the value of well trained, capable men. The fact that few plants only have established methods of training workmen does not necessarily mean that the managers are satisfied with that condition, but rather that they know of no training system that can be satisfactorily operated in their plants, and as questions are sure to be asked about the method of introducing this system, a few words on that subject may not be amiss, it being borne in mind that *the changing of a system of management is a very serious matter, and cannot be done by a busy superintendent in his spare time.*

METHOD OF INTRODUCTION

48 In order to set tasks we must know beforehand what work is to be done and who is to do it. In order to pay a bonus, we must know after the work is done whether it was done exactly as specified. Hence our first care in starting to introduce this method is to provide means for assigning tasks to the workmen, and means for obtaining such a

complete set of returns as will show just what each man has done. When this much has been introduced, the output of a plant is always increased and the cost of manufacture reduced.

49 The next step is to separate such of the work as is standard, or can be readily made standard, from the more miscellaneous work, and set tasks for the standard work. Then we begin to standardize, and as fast as possible reduce the expert and increase the routine work. The effort to classify and standardize expert knowledge is most helpful to the experts themselves, and in a short time they begin to realize that they can use their knowledge far more efficiently than they ever dreamed.

50 As soon as work has been standardized, it can be intelligently planned and scheduled, each workman being given his specific task, for which he is paid a bonus when it is done in the manner and time specified. As bonus is paid only on the written statement of the inspector that the whole task has been properly done, failure to earn a bonus indicates that our plans have not been carried out.

51 An investigation of every case of lost bonus keeps the management closely in touch with the progress of the work, and as the workmen are ever ready to help disclose and remove the obstacles that prevent their earning their bonus, the managing problem is greatly simplified; for, as one of my co-workers has very aptly put it, "*the frictional lag due to the inertia of the workman is changed by the bonus into an acceleration.*"

52 With increase in the number of bonus workers, this force of acceleration increases, and not only does the careless worker, who by his bad work prevents some other from earning his bonus, fall into disfavor, but the foreman or superintendent who is lax in his duty finds his short-comings constantly brought before him by the man whose duty it is to investigate all cases of lost bonus.

MORAL TRAINING

53 The fact that under this system, everybody, high and low, is forced by his co-workers to do his duty, for some one else always suffers when he fails, acts as a strong moral tonic to the community, and many whose ideas of truth and honesty are vague find habits of truth and honesty forced upon them. This is the case with those in high authority as well as those in humble positions, and the man highest in authority finds that he also must conform to laws, if he wishes the proper coöperation of those under him.

FUEL ECONOMY TESTS AT A LARGE OIL BURNING ELECTRIC POWER PLANT HAVING STEAM ENGINE PRIME MOVERS

By C. R. Weymouth, San Francisco, Cal.

Member of the Society

The writer desires to present to the Society the results of various fuel economy tests at the Redondo Plant of the Pacific Light and Power Company, near Los Angeles, Cal., which are of interest both by reason of steam engines having been used as prime movers and because of the notable economy attained. Owing to the rigorous demands on commercially operating power stations, an opportunity is seldom afforded to conduct uniform load tests on a complete plant unit, and the information secured is therefore considered worthy of presentation before the Society.

2 No doubt many members have noted in September issues of some of the leading technical papers a description of the apparatus constituting the Redondo plant. For the better understanding of the results herein presented, a brief statement of the plant make-up will be given.

PLANT DESCRIPTION

3 The station consists of three main units of 5000 kw. each; the layout being in general on the panel system. For each plant unit, there is one McIntosh and Seymour double horizontal and vertical, compound, condensing, automatic, grid-iron valve engine, size 34 and 70 by 56 in., directly connected to an alternator. The rated speed is 100 r.p.m. Each engine has two horizontal high pressure steam cylinders, and two vertical low pressure cylinders and is designed to operate at 175 lb. maintained throttle pressure, with 100

[Data upon California oils and their heat values will be found in the paper by the author upon "Unnecessary Losses in Firing Fuel Oil and an Automatic System for Eliminating Them," to be presented at the Annual meeting, December 1908.—Editor.]

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deg. fahr. superheat at the throttle. All steam cut-off valves are under control of the one shaft governor. This engine governor is subject to variation in speed, under the control of a McIntosh and Seymour electrically operated speed-changing mechanism, situated within the governor and operated from the switchboard gallery. Each engine is fitted with two superheating receivers designed not only to separate the moisture and oil in the exhaust from the high pressure cylinder, but also to superheat the steam en route to the low pressure cylinders.

4 Directly connected to each engine there is one ATB-60-pole, 5000-kw. 100-r.p.m., 18 000-volt, 50-cycle, 3-phase General Electric generator of the flywheel type. The revolving field is provided with squirrel cage winding to prevent hunting. The generator field rheostats are motor-operated with remote control from the switchboard.

5 In each plant unit are six Babcock and Wilcox boilers of forged steel construction, arranged in three batteries of two each. Five boilers are intended for the nominal capacity of the unit, the sixth boiler for reserve. The boilers are 21 sections wide, 14 tubes high, 18 ft. long, with three 42-in. drums, each containing a total effective water-heating surface of 6042 sq. ft. and designed for 200 lb. working steam pressure. Each boiler is equipped with one Babcock and Wilcox forged steel double-loop superheater.

6 The boilers are provided with Peabody patent fuel oil burning furnaces, having a furnace depth of 10 ft. from boiler front to face of bridge wall. The burner head is placed at the bridge wall, the flame shooting toward the boiler front. The furnace diverges in the direction of the flame, corresponding to the angle of inclination of boiler tubes. Three burners are used per boiler, and are controlled from the boiler front.

7 Corresponding to each main generator, there is one 75-kw. General Electric exciter, driven by means of a 9 and 17 by 12 in. Harrisburg tandem, compound, non-condensing, piston valve, automatic, enclosed, self-oiling engine.

8 There are two Wheeler condensers for each main engine, located in the basement below the operating floor, one condenser being used for each low pressure cylinder. The exhaust steam is led directly from the low pressure cylinder into a 30-in. exhaust main, and after passing through a 90-deg. sweep, drops directly into the condenser. The two condensers on each engine are cross-connected so that both engines can exhaust into either condenser when the other is undergoing cleaning or repairs. Each condenser contains 5002 sq. ft. of

cooling surface, consisting of $\frac{3}{4}$-in. O. D. No. 18 B.W.G. brass tubes. In addition to the regular cooling surface of each condenser, there is provided in the upper compartment a series of tubes comprising a Volz Heater, through which the condensed water is pumped by the air pump at atmospheric pressure. The feed water is thus heated to within a few degrees of the temperature of the exhaust steam surrounding the upper tubes, thus compensating for the cooling action due to the lower condenser tubes.

9 Circulating water is supplied by three large engine-driven centrifugal pumps, connected to a common pump discharge main. Branches from this main supply cooling water to all condensers. The outlets from all condensers connect to a common condenser discharge main. Each condenser has one Edwards triplex, suction, valveless, single action, motor-driven air pump, size 16 by 10 in.

10 After passing through the Volz Heaters the condensed water is passed through suitably constructed filters for the elimination of the greater part of the entrained cylinder oil. The outlet of the filters is connected to the feed pump suction main, the supply of water being maintained by an open equalizing hot well, which is also connected to the suction main.

11 Each unit has one Snow duplex, horizontal, boiler-feed pump, having a compound non-condensing steam end and an outside center-packed water end. Each unit also contains one Goubert vertical, closed, auxiliary, feed water heater, of the multiple-flow type, having 1000 sq. ft. of effective tube-heating surface.

12 The condensation in the main engine superheating receiver coils, is led to a simple duplex receiver pump. The hot water, at a temperature of about 360 deg., is returned under boiler pressure to one or more of the boilers in the corresponding unit. The oily drips from receiver bodies are trapped to waste. From the auxiliary measuring tanks, the fuel oil is first pumped through a Goubert closed, multi-flow oil heater, in which the oil is heated to a temperature of approximately 150 deg. fahr., utilizing the exhaust steam from the oil feed pump. From the oil heater the oil is led to an oil pressure main and thence to the oil burners through a suitable system of piping.

13 An automatic system of regulation was employed for the firing of all boilers in the unit tested, hand firing being used on the remainder of the plant. The automatic system controls the supply of oil to the burners, the supply of steam for atomizing purposes, and the supply of air for combustion. This control is obtained through a steam pressure regulator which operates a relief valve in the oil

pump discharge line, and is actuated by the variations in steam pressure in the boiler. This means of control causes a variation in the rate of burning oil comparable with the momentary load. With this system all burner valves may be left wide open or nearly so, and it follows that the intensity of firing increases or decreases simultaneously in all the boilers.

14 The supply of steam to burners for atomizing purposes is from a low pressure steam main, the controlling valves at each of the burners being left wide open or nearly so. The variations in pressure in the low pressure steam main are governed automatically by variations of the oil pressure in the oil main. The regulator used throttles the supply of live steam to the low pressure main, to produce the desired relationship of oil pressure and steam pressure on the respective burner headers. The supply of air for combustion is automatically regulated by a damper controller, operating a common rock shaft connected to all boiler-outlet dampers. This regulator opens and closes all boiler dampers synchronously with an increase or decrease of oil pressure in the oil main.

15 This system is capable of such operation as to control all of its functions completely and automatically. In actual practice, however, it has been found desirable to watch closely the firing of boilers to make certain that no external conditions such as clogging of burners, distortion of flames by burners, variation in temperature of fuel oil as fired, etc., are present to prevent the attainment of the highest possible efficiency. To minimize heat losses due to radiation, all live steam heat-radiating surfaces are covered with non-conducting covering 3 in. in thickness, and other surfaces are covered in accordance with the dictates of good practice.

OUTLINE OF TESTS

16 The data and results given herein are based on the following tests:

a Official 15-day variable load test on No. 2 plant unit.

b Uniform load test at approximately 2000 kw. output No. 2 plant unit.

c Uniform load test at approximately 3000 kw. on No. 2 unit.

d Uniform load test at approximately 4000 kw. output No. 2 unit.

e Uniform load test at approximately 5000 kw. output No. 2 unit.

f Test complete plant at variable load similar to the official test.

These tests are more fully described under their respective headings.

OFFICIAL FIFTEEN-DAY TEST

17 The official test of this station was on No. 2 plant unit, and was made to determine the economy of the plant under contract conditions, as a basis for computing the bonus earned by the contractor, or the penalty due the owner. The result of this official test is now common knowledge, but not all the conditions surrounding the test have been heretofore published; and as the result of the official test on variable load is of interest for comparison with the results of the uniform load test, herein presented, the writer desires to review the more important conditions attending the official trial. The contract placed the official test under control of a testing committee, the members selected being Prof. C. L. Cory, Dean of the College of Electrical Engineering, University of California, as Chairman; Edward S. Cobb, Chief Engineer for the Pacific Light and Power Company, as Purchaser's Member; and the writer, having charge of the engineering department of Chas. C. Moore and Company, Engineers, as Member for the Contractor (or Company, as herein referred to in extracts from the contract).

18 The Contractor's commercial load economy guarantee was as follows:

> The Company guarantees * * * that the "First Unit" * * * when operating on commercial railway load, under 90 days test, conforming to the conditions herein specified, will develop an average economy of 170 kw-hr. per barrel of oil.
>
> Duration of test will be 90 days.
>
> The total net output of Unit under test for each day will not be less than 60 000 kw-hr., and not more than 78 000 kw-hr.
>
> The main Unit under test will be in operation during 19½ consecutive hours per day, and will be shut down 4½ hours per day. The temperature of circulating water entering the condensers will never exceed 70 deg. fahr.
>
> The power factor of the electric load will be at all times between 80 and 100 per cent. Total load on 5000-kw. alternator will never exceed 6000 kw. The load on 5000-kw. generator greater than 5000 kw. will last for an interval not greater than ½ min. duration; and there will be not to exceed 10 such intervals during any hour that the plant is under test.
>
> The load on the Unit under test will be such as would be produced by the regu-

lar and normal working of the railway system to which the power is furnished by the purchaser, and varying within the limits herein described.

The load will not be manipulated so as to be either favorable or unfavorable to the showing of Unit under test, within the limits herein specified.

19 It was optional with the contractor to insist on the operation of the test unit on a separate transmission line. This would have inconvenienced the purchaser, and for operating reasons a compromise was effected, it being agreed to operate the test unit during the official test in accordance with an agreed load curve, subject to certain upper and lower limits as shown in Fig. 1, in which:

Curve A shows agreed load curve to be followed as closely as possible by electrical operator, as determined by testing committee.

Curve B shows agreed maximum load not to be exceeded by unit during test.

Curve C shows agreed minimum load curve below which the load was not to fall during test.

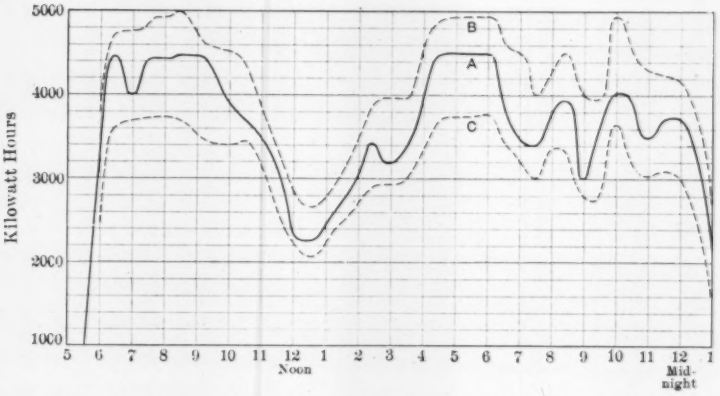


FIG. 1 AGREED LOAD CURVES 5000-KW. UNIT OFFICIAL TEST

CURVE A SHOWS AGREED LOAD CURVE TO BE FOLLOWED AS CLOSELY AS POSSIBLE BY ELECTRICAL OPERATOR AS DETERMINED BY TESTING COMMITTEE; TOTALING 72 280 KW-HR. DAILY OUTPUT. CURVE B SHOWS AGREED MAXIMUM LOAD NOT TO BE EXCEEDED BY UNIT DURING TEST. CURVE C SHOWS AGREED MINIMUM LOAD OR LOWER LIMIT OF OPERATION DURING TEST.

20 The control of the load during test was thus left entirely to the purchaser's station operator within the specified limits and subject to such momentary variations as would be imposed on the test unit by the fluctuation in load on purchaser's system. It was thus

attempted to generate a total of 72 280 kw-hr. per day, with the understanding that should the total kilowatt-hour output up to any hour exceed that shown by the load curve A, then during the remainder of the test period the load was to be modified to correct for any variation in the total output, so as to maintain the total daily output of the unit uniform and as agreed.

21 Endurance features having been established, by reason of the long period of operation previous to the test, the period of test was subsequently reduced from 90 to 15 days. And owing to certain advantages that might accrue to the contractor, such as superior economy due to shortening of test, the contractor agreed to deduct from the bonus otherwise due the sum of $50 000 in consideration of this reduction of test period.

22 Owing to the use of a circulating water discharge main, common to all circulating water pumps, and the steam supply pipes of the circulating pump engines being from a common steam main, it became necessary to eliminate the centrifugal circulating water pump from the unit under test; and in consideration of this omission the contractor rebated the purchaser an agreed amount of the bonus.

23 It was further agreed that the amount of circulating water taken by the two condensers, for the unit under test, should correspond to a rise in temperature of circulating water within the condenser, of between 15 and 18 deg. fahr., such amount of water being controlled by the circulating water outlet valves.

24 In connection with the guarantees, however, it should be stated that the economy guarantee was based on the purchaser's supplying such a siphon system of suction and return circulating water piping, that the total head imposed on circulating pump due to friction and flow of water within pipes, would not exceed a total of 5 ft.

25 The economy guaranteed initially included the provision for lighting the unit under test from energy generated by the test unit. Correction for lighting was made by deducting from the total net output of the unit under test an amount of power equivalent to 125 kw-hr. per day, and the reported economy is based on the net figure after making this deduction.

26 During the test the power for operating the air pump motors, for operating motors connected to circulating pumps used for passing cooling water through engine bearings and guides, for operating electric motor in main engine speed-changing device, and for operating electric motor for controlling switches, was taken from the leads

of the main generator in such a way as to compel the watt meters to indicate only the net useful output of the unit.

27 It is only fair to state that, owing to the layout of the plant, certain auxiliary apparatus in connection with the plant was installed by the purchaser, power for which was not taken from the unit under test; such auxiliaries include the following:

a Electric motor for pumping oil from the main oil storage tank to weighing tanks: the purchaser was compelled to install this electric pump owing to the topography of plant site, the usual gravity feed from main tanks to auxiliary suction tanks being impossible, by reason of the elevation of temporary receiving tanks on weighing scales.

b Electric motor for pumping feed water from the wells, constituting the purchaser's fresh water supply plant, and delivery to the plant under a slight pressure, equivalent to a normal city supply.

c Two small Westinghouse air compressors common to the entire plant, for compressing air, used for cleaning generators, operating certain switches, and forcing cylinder oil from the filtering tanks through a piping system, to save labor of handling by oilers.

d Sump pump common to the entire plant, necessary for pumping overboard waste water from the various sumps; these pumps were installed by the purchaser by reason of the level of sump being lower than sewer.

e Electric motor of 50 h.p. for operating 12 by 12 in. duplex vacuum pump, installed by purchaser to remove accumulation of air in long suction and return circulating water pipes; this motor operated intermittently.

It will be recognized that the auxiliaries omitted, with the exception of the circulating pumps, are not necessarily essential in the guarantee of power, that is, they may or may not be required in a power plant, depending upon local conditions, etc.

28 During the tests special weighing and receiving tanks were provided for the measurement of the fuel oil required by the unit under test. After weighing, the oil was emptied into a receiving and heating tank, the heating being done by steam from the unit under test, and from there conveyed by gravity to the oil feed pump used for feeding oil to the burners.

29 All instruments were accurately calibrated and standardized. For purposes of the tests a total of twelve integrating watt meters were purchased, and after calibration in place under conditions of operation, it was finally decided to determine the power output by using one watt meter in each phase of the generator, there being three reserve watt meters. Meters were connected in the main winding of the generator between the grounded center of the winding and the winding itself.

30 For the calibration of electrical instruments the testing committee employed Mr. E. F. Scattergood of Los Angeles, an Electrical Engineer and a specialist in such work. There were used certain standards furnished by him, certain standards supplied by the electrical engineering department of the University of California, and certain other standards secured from the Bureau of Standards, Washington, D. C. The oil analyses and heat determinations were made by Prof. Edmond O'Neill, of the department of chemistry, University of California. Check analyses and determinations were made by other chemists.

31 A large corps of assistants and observers were employed during the tests. At one time a total of 80 persons was directly involved. The neutral observers, including the members of the graduating classes in Electrical and Mechanical Engineering of the University of California, were under the direction of the chairman of the testing committee. In all details of the test care and accuracy, in keeping with the importance of the desired results and the amount of bonus money involved, were observed.

UNIFORM LOAD TESTS

32 During the uniform load tests the same general conditions prevailed as during the official trial, except as herein indicated. Though unofficial, the determination of fuel burned and power output during the uniform load tests was still under control of the testing committee and the same corps of observers, although the watt meter readings were taken less frequently, as a result of experience during the official trial. The calculated economies given are obtained from the readings made by the neutral observers. It should be stated however that the testing committee never completed any official report for these uniform load tests, as the results of the uniform load tests were of no consideration in determining the amount of bonus to be paid the contractor.

VARIABLE LOAD TEST OF THE COMPLETE PLANT

33 During this test the entire plant was operated under commercial conditions of load, except that an attempt was made to follow on all three units the load curve used during the official test. The circulating pumps were supplied with steam from the various boilers, and all plant auxiliaries, including those omitted from official test, were included in the complete plant test. All these last named auxiliaries were not in use at all times, however, but only for the following periods: 10-h.p. motor for water supply pump, in operation during, entire test; 50-h.p. motor for vacuum pump, 16½ hr.; and 10-h.p. motor-operating pump for pumping oil from main to auxiliary tanks, in use during greater part of test. The output of all units was determined by means of station watt meters which have since been carefully calibrated by the purchaser, and the correction factors as found applied to the observed readings.

34 This test did not in any sense do justice to the plant, owing to operating conditions as explained below, and for this reason the test was protested by the writer and at the time was discarded by the testing committee. The writer has since felt, however, that the result is of interest, and offers it with this explanation.

OPERATING CONDITIONS DURING TESTS

35 During all the tests the plant was handled by the regular station operators under the control of the contractor's superintendent of construction, Mr. J. R. Atchison.

36 The load was quite variable, necessitating constant adjustment of auxiliaries, oil burners, dampers, etc. On boilers of No. 2 unit, the automatic system of firing was used during the official test and during uniform load tests. Control of oil burners and dampers on No. 1 and No. 3 units during the complete plant test was by hand.

37 The contractor's experts were present during all tests and cautioned the operators when necessary. Dr. D. S. Jacobus, a member of the Society and Advisory Engineer for the Babcock and Wilcox Company, was present during all the tests, and as manufacturer's representative; Mr. Harte Cooke, a member of the Society, representing McIntosh, Seymour and Company, was also present during all the tests, and much credit is due to both for the exceptional performance of representative apparatus.

38 During the official 15-day test certain unfavorable conditions predominated, for which no correction has been made in the stated

economy. Owing to the accumulation of sea weed in the circulating water pipe line and the interruption of flow of circulating water through the condensers, there were intervals when either one or both condensers of the test unit operated at a reduced vacuum, and at other times it was necessary to run the engine non-condensing. Frequent shorts occurred on the lines, and again periods of excessive overload. During the official test, instead of following the comparatively smooth load curve agreed, as shown in Fig. 1, momentary changes in load were of considerable magnitude. Fig. 2, curve A, shows the actual load curve reproduced from the record of the Westinghouse polyphase, curve-tracing, indicating, station watt meter, for one day of the 15-day test, being an average day in regard to the momentary variations of load. Actual load curves are similarly given for all uniform load tests. A marked saw-tooth effect will be noted, momentary changes being so violent as to cause unfavorable economy, these variations in load not only affecting the economy of the prime mover, but also the performance of boilers, the continual change in rate of oil firing, air supply, steam supply to burners, etc., making it impossible to secure the best results; although the automatic firing system employed on No. 2 unit no doubt greatly reduced such losses.

39 It will be noted that certain of the uniform load tests were of comparatively short duration. The results of such tests are therefore subject to slight error. The initial period of starting the uniform load tests was determined by the testing committee, and while the writer has reluctantly assented to certain of these intervals, it will be self-evident that for certain of these tests the period allowed for warming up boilers before starting of test was not sufficient to eliminate all of the heat storage effect, and is not fully in accordance with the Society's rules for conducting such trials. Previous to certain tests, the boilers had been down for such lengths of time as to be practically cold. In other cases the boilers had been at stand-by for four or more hours previous to starting fires; except during the 3000-kw. test where the boilers after previous day's run were kept up to steam pressure by intermittent firing. The results, therefore, from this standpoint are slightly unfavorable, the fuel consumption stated involving to some extent the fuel loss due to warming up the boilers.

40 The test for the 2000-kw. load was made with three instead of five boilers in operation, as is the regular practice at this plant during any prolonged running at such loads. Owing to the short period of the test, it would have been impossible to keep the two idle boilers on the line and fire at intervals during the test, in such a manner as to

guarantee the same amount of heat storage in the idle boilers at both commencement and end of the test period. The amount of such loss due to stand-by of the idle boilers was therefore determined separately by actual measurement, and the resulting fuel loss is given in the accompanying table, corresponding to which due correction in economy has also been figured. The amount of this correction corresponds to the heat necessary to keep the two idle boilers continuously up to full boiler pressure. It is a fact, however, that the actual stand-by loss was less than this amount, owing to the reduction in the rate of heat radiation which takes place after the boiler has appreciably cooled.

41 During the uniform load tests it was necessary, owing to peculiar conditions of operation, to lower materially the power factor of the unit under test, in order to control the load on same more readily. Owing to this lowering of the power factor below the range for which the watt meters were carefully calibrated, an error was introduced in the readings, manifesting itself in a tendency of the meters to run too slow, thus indicating an output less than actual. Further, owing to the lower power factor imposed on the main generator, the C^2R and other losses were greater than normal (80 to 100 per cent), causing a falling off in generator efficiency and consequent reduction of fuel economy.

42 During the complete plant test the head on the circulating pumps was measured and averaged in the neighborhood of 35 ft. This excess head is due to the fact that the purchaser met with certain reverses in the installation of the circulating water pipe line, making it necessary to install internal-angle iron strengthening ribs. The suction and discharge piping is also subject to certain air leakage, and to a considerable accumulation of sea weed, both in the main suction line and in the suction line strainers. As a result of this condition, not only were the total head and duty imposed on the circulating pumps during the complete plant test exorbitant, but the quantity of circulating water handled was not quite sufficient for the most economical vacuum requirements when all three units were in operation.

43 For periods during the complete plant test, the vacuum was very poor indeed, owing to the accumulated sea weed in the suction line and condensers faster than the strainers could care for it. During all these tests, certain unfavorable conditions existed as to the character of oil burned and the treatment of same preparatory to burning, as explained more fully below under the caption, Fuel Oil Specifications and Corrections.

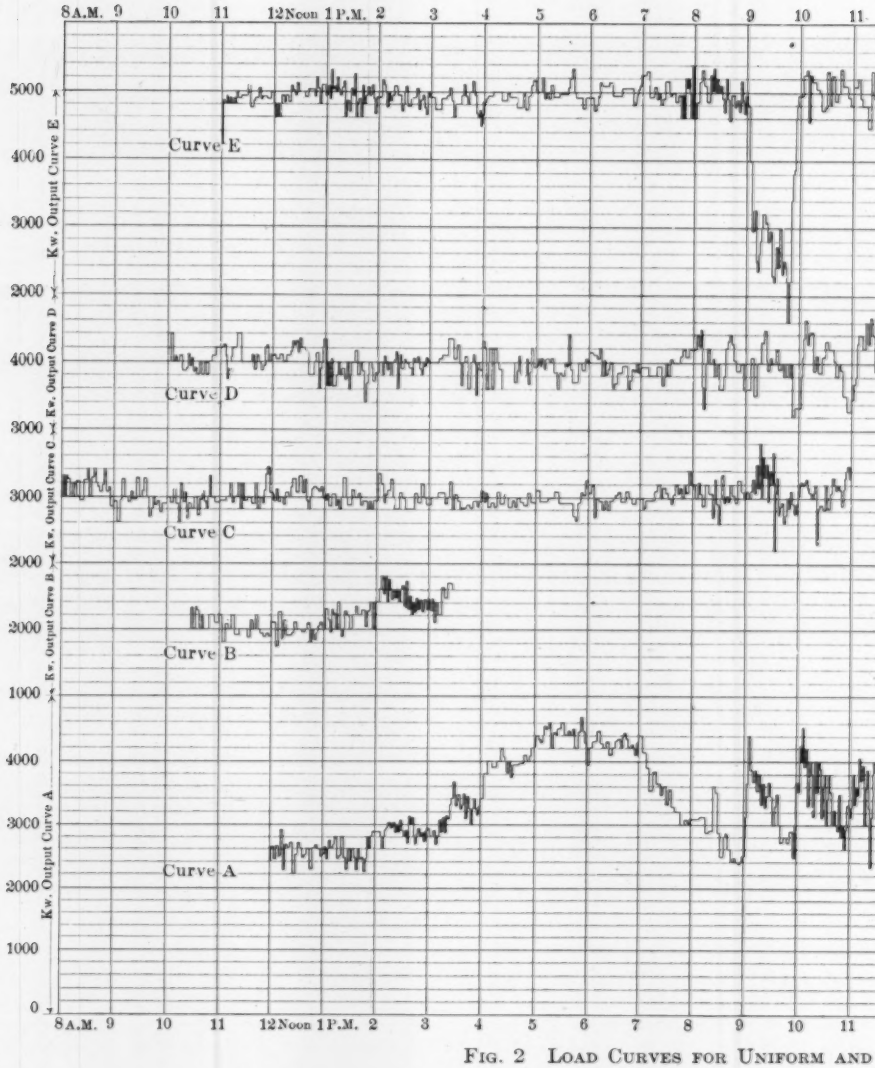
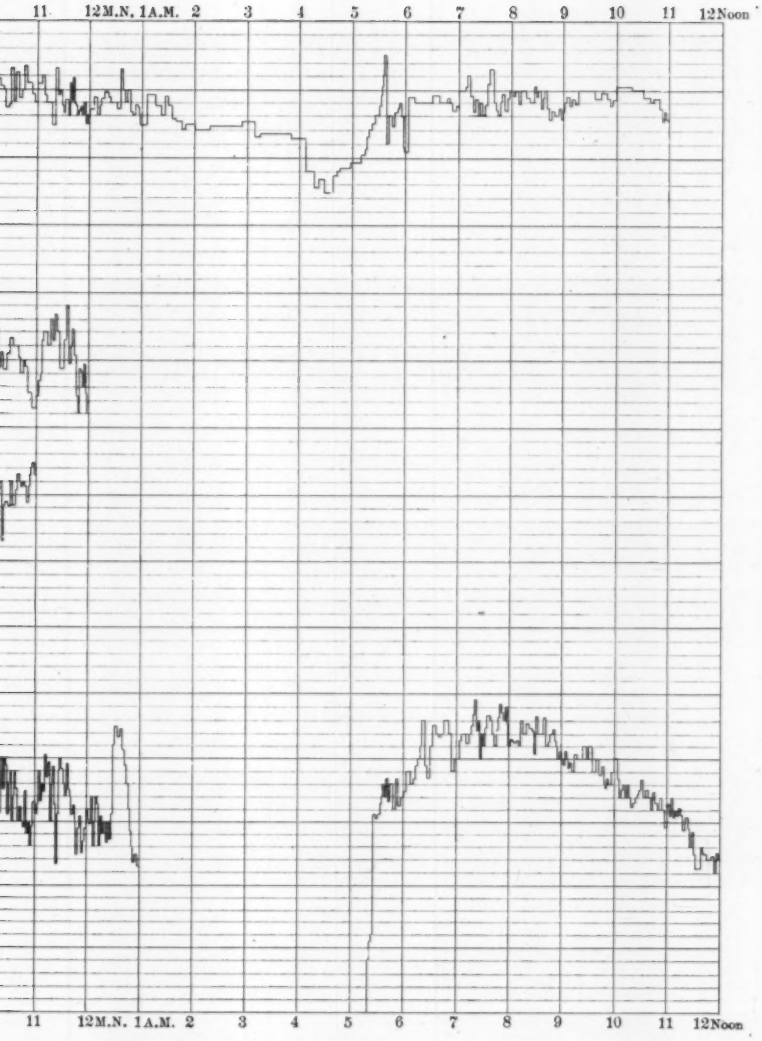


FIG. 2 LOAD CURVES FOR UNIFORM AND

CURVE A SHOWS THE LOAD VARIATIONS FOR THE ELEVENTH DAY OF THE 15-DAY TESTS, 12 NOON TO 12 NOO
TEST, 10.30 A.M. TO 3.30 P.M. CURVE C SHOWS THE LOAD VARIATIONS FOR THE TEST AT APPROXIMATEL
AT APPROXIMATELY 4000-KW., 10 A.M. to 12 M. CURVE E SHOWS THE LOAD VARIATIONS FOR THE TEST



M AND VARIABLE LOAD TESTS

12 NOON. CURVE B SHOWS THE LOAD VARIATIONS FOR THE APPROXIMATE 2000-KW. LOAD
IMATELY 3000-KW., 8 A.M. TO 11 P.M. CURVE D SHOWS THE LOAD VARIATIONS FOR THE TEST
HE TEST AT APPROXIMATELY 5000-KW., 12 NOON TO 11 A.M.

OBSERVED DATA

44 In Table 1 is given a summary of all the observed data and the calculated results for the various tests, including averages of steam pressure at engine throttles, superheat at engine throttles, condenser vacuums, circulating water temperatures, etc.

FUEL OIL SPECIFICATIONS AND CORRECTIONS

45 Certain arbitrary corrections were agreed on in formulating the guarantee of this plant, and the nature of such corrections should be well understood in order to gauge the value of the results herein given and the possible error that may have been introduced by reason of such corrections.

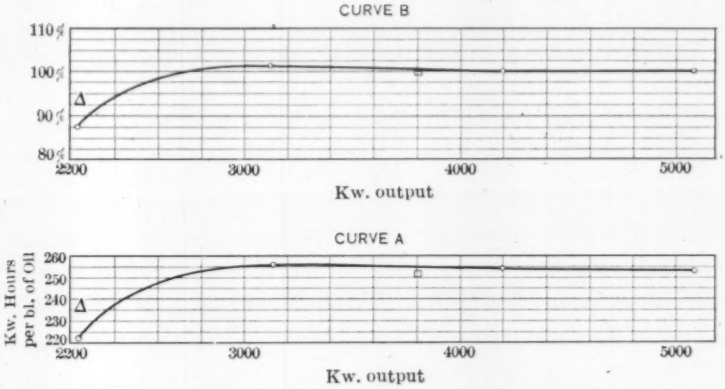


FIG. 3 CURVES SHOWING ACTUAL AND RELATIVE FUEL ECONOMY AT FRACTIONAL LOADS, 5000-KW. UNIT

CURVE A SHOWS FUEL ECONOMY AT VARIOUS FRACTIONAL LOADS IN KILOWATT HOURS PER BARREL OF OIL. CURVE B SHOWS RELATIVE FUEL ECONOMY AT FRACTIONAL LOADS AS COMPARED WITH ECONOMY AT RATED LOAD OF 5000 KW. POINTS MARKED □ WERE OBTAINED FROM THE AVERAGED LOAD OF THE 15-DAY VARIABLE LOAD TEST. POINTS FOR THE 2000-KW. TEST WERE CALCULATED FOR 5 BOILERS IN SERVICE INSTEAD OF 3 AS ACTUALLY USED. POINTS FOR 3 BOILERS SHOWN THUS △.

46 It is well known that with the poorer grades of coal, having excessive quantities of ash, sulphur, moisture, and other volatile matter, there is a falling off in the furnace efficiency as compared with the combustion of the better grades of coal; so with crude oil having excessive quantities of moisture, sulphur and silt, or of low specific gravity, there is an appreciable falling off in the furnace efficiency as compared with a grade of crude oil of average quality. It is certain however that most of the losses due to foreign matter are less with oil fuel than with coal, but the exact extent of these losses is so far unknown.

TABLE 1 SHOWING RESULTS OF VARIABLE AND UNIFORM LOAD TESTS ON ONE 5000-KW. UNIT, ALSO VARIABLE LOAD TEST ON COMPLETE PLANT OF PACIFIC LIGHT AND POWER COMPANY, REDONDO, CAL.

1	Designation of test	Unit	Variable load average of 15-day tests	2000-kw. load test (Approximate)	3000-kw. load test (Approximate)	4000-kw. load test (Approximate)	5000-kw. load test (Approximate)	Variable load complete plant test
2	Date of starting test	1908	April 18	May 14, 10:30	May 19, 8 a.m.	May 18, 10 a.m.	May 5, 12 noon	May 21, 11 a.m.
3	Date of stopping test		May 4	May 14, 3:30	May 19, 11 p.m.	May 18, 12 m. n.	May 5, 11 a.m.	May 22, 11 a.m.
4	Duration of test	Hours	24	5	13	14	23	24
5	Time of starting fires			5:10 a.m.	3:00 a.m.	5:10 a.m.	4:35 a.m.	5:05 a.m.
6	Period of warming boilers	Hours		5:20	5:0	4:50	7:25	5:55
7	Average steam pressure at engine throttles	Lb. per sq.in.	180.03	183.	181.74	180.9	189.3	173.76
8	Average superheat at engine throttles	Deg. F.	82.46	87.99	92.	96.05	92.78	95.15
9	Average temperature circulating water inlet	Deg. F.	63.03	61.64	61.41	61.34	62.4	59.33
10	Average temperature circulating water outlet	Deg. F.	79.15	78.09	79.37	79.02	81.09	82.12
11	Average vacuum in condenser (Corresponding 30 in. Bar)	In. Hg.	28.334	28.426	28.343	28.214	27.976	27.784
12	Average temperature of feed water leaving heater	Deg. F.	146.9	184.3	167.8	155.9	150.7	177.22
13	Kilowatt output (including lights)	Kw-hr.	71615.24	11225.577	47126.457	58745.125	116899.748	215262.438
14	Deduction for lights	Kw-hr.	125.	26.042	78.125	79.917	117.792	.....
15	Net kilowatt output, deducting lights	Kw-hr.	71490.24	11199.535	47048.332	58665.208	116781.956	.....

TABLE I.—*Continued.*

16	Fuel oil as fired (334 lb. to bbl.)	Bbl.	303.387	50.01	195.458	244.783	496.910	957.566
17	Heat units per pound oil as fired	B.t.u.	17840.	17938.8	17920.8	17965.8	17838.	17717.
18	Sulphur in oil (by weight)	Per cent	2.34	2.17	2.43	2.39	2.49	2.60
19	Moisture in oil (by weight)	Per cent	2.38	1.82	2.08	1.895	2.70	2.59
20	Silt in oil (by weight)	Per cent	.14	.138	.14	.113	.10	.....
21	Fuel oil corrected as per contract	Bbl.	282.746	47.219	183.307	230.764	460.884	883.115
22	Economy (oil corrected as per contract)	Kw-hr. per Bbl.	252.842	237.298	256.664	254.252	253.382	243.758
23	Economy (oil corrected as per contract)	B.t.u. per Kw-hr.	24438.	26039.	24074.	24302.	24386.	25349.
24	Economy corrected only for heat units in oil	B.t.u. per Kw-hr.	25288.	26742.	24857.	25027.	25347.	26320.
25	Number of boilers in service	No.	5	3	5	5	6	15
25	Additional fuel oil 2000-kilowatt test account idle boilers equivalent 5 boilers in use	Bbl.	.....	50.339	.....	.....	.....	.....
27	Calculated economy 2000-kilowatt test equivalent 5 boilers in service (oil cor.)	Kw-hr. per Bbl.	.....	222.482	.....	.....	.....	.....
28	Combined efficiency of engine and generator based on separate exciter	Per cent	.....	90.2	92.5	94.1	94.75	.....

47 The writer would explain that during the preliminary negotiations for the Redondo plant the contractor's guarantee of 170 hw-kr. per barrel of oil was based on an average quality of Bakersfield crude oil, of known furnace performance, having the following specifications:

Specific gravity, 16 deg. Baume.
Corresponding weight per bbl., 336 lb.
Heat units per lb. of oil calorimeter test, 18 500.
Moisture limit, 1 per cent.
Sulphur limit ½ per cent.

48 On the day of closing the contract the purchaser insisted on testing the plant with oil secured from the purchaser's nearby wells; and while this oil never had been used during any authentic boiler trials, yet it was known to be of low gravity, and low in heat units, to have considerable quantities of both moisture and sulphur, and in general to be a decidedly inferior grade of oil.

49 After some discussion a compromise was affected, whereby the economy guarantee was allowed to remain as stated, the oil specifications being changed to the following:

Minimum specific gravity of oil, 14 deg. Baume.
Maximum specific gravity of oil, 18 deg. Baume.
Agreed weight per bbl., 334 lb.
Minimum heat units per lb. of oil, 17 000.
Moisture limit, 5 per cent.
Sulphur limit, 2½ per cent.
Silt, etc., limit, ½ per cent.

50 It was finally agreed that in addition to correcting on the heat unit basis a further correction should be made for the presence of sulphur exceeding the limit of 1 per cent or moisture exceeding ½ per cent and the purchaser should be penalized by deducting as oil from the weight of oil fired the weight of all moisture in excess of ½ per cent, all sulphur in excess of 1 per cent and all silt and other foreign matter.

51 It is a well known fact that oil, low in gravity, requires an excess of steam for preheating in the auxiliary suction tanks, both for the preliminary treatment of the oil in an attempt to remove the excess of moisture and silt by warming and also to enable the oil pressure pumps to lift this oil, under suction without danger of interruption in the pumping of oil to the burners. It is also well known

that all crude oils are complex mixtures of various hydro-carbons, that distillation takes place to varying extents at all temperatures, and that the higher the temperature to which the oil is heated in an open tank, the greater the loss due to such distillation. The writer had tests made by Mr. C. H. Shepard, chemist, of San Francisco, to determine as near as may be the loss due to distillation caused by such preheating in the case of the Redondo tests and his report is as follows:

Regarding the loss by distillation in open tank at temperatures of 120 deg. fahr. and 130 deg. fahr., you stated in your letter that you were using about 100 000 lb. of oil per day and that the diameter of your tank is about 10 ft. I assume that you would have about 40 000 lb. of oil in the tanks. If the oil were flowing in and out at the same rate any particle of oil would take at least eight hours to pass through the tank. I have kept one sample for eight hours at 120 deg. fahr. and another for eight hours at 130 deg. fahr. in open vessels 4 in. in diameter and oil to a depth of about 3¼ in. without agitation. Results are as follows:

TABLE 2

	At 120 deg. fahr. for 8 hr.	At 130 deg. fahr. for 8 hr.
Total loss in weight	1.53%	1.95%
Loss of water	.35%	.59%
Loss of oil	1.18%	1.36%
Heat unit loss at 20 000 per lb.	.236 B.t.u.	.272 B.t.u.

Per cent in water in original sample was by distillation 2.75 per cent.

52 It is further well known that the heavier the oil, the greater the amount of heating necessary in the pressure heaters between pumps and burners, where heat is added to facilitate atomization of the oil and to economize in the amount of steam necessary for such atomization. In addition to the general falling off in furnace efficiency due to the poorer grade of crude oil, there was a further and rather indeterminate loss due to the effect of sulphur and silt, which caused a slight clogging and corrosion of the burners and at times a distortion of the flames. The sulphur fumes from the products of combustion also escaped from the boiler setting, causing a serious annoyance to the firemen and necessitating an excess of air supply to overcome the trouble. With the particular setting of front and rear dampers for greatest economy and minimum air supply, there was at the top of the first pass inside of the boiler setting, a positive pressure in lieu of a draft, due to the lighter column of very high temperature

gases within the boiler setting, as compared with the weight of the outside air at the boiler room temperature. Sulphur was present in the oil to such an extent that the firemen were seriously annoyed unless an excess of air was admitted, increasing the draft within the boiler setting and preventing the escape of sulphurous fumes, thereby causing the well recognized loss in economy due to excess air.

53 With our present knowledge of the combustion of oil fuel, it is impossible to rationalize the results of the Redondo tests so as to determine exactly the true thermal efficiency that would have been attained had an average grade of crude oil been used. During the tests, as will be noted from the values given in Table 1, the grade of oil used was of very low specific gravity and contained excessive quantities of moisture, sulphur and silt. It is indisputable that an appreciable loss in efficiency resulted from the presence of these foreign substances, but owing to the arbitrary nature of the corrections made the plant economies herein stated may be in error by small percentages, the amount of which must remain indeterminate. This criticism will likewise apply to all similar tests which are not made with a grade of fuel recognized as standard.

54 As a matter of interest, the writer has included in Table 1 figures showing the plant efficiency omitting the contract corrections; it must be borne in mind, however, as above stated, that these figures evidently represent a lower economy than would have been obtained with the standard grade of oil specified.

55 Considerably better economy would be expected in the uniform load tests than was secured in the official test, namely 252.842 kw-hr. per barrel of oil. The actual results were not very much higher, however, for as already explained under the heading of Operating Conditions During Tests, the period allowed for warming up the boilers before starting certain of the uniform load tests was not sufficient to overcome the loss due to radiation during the lay-over period and the results were lower than they should be on this account. It should also be borne in mind that in the so-called uniform load tests the actual load varied from time to time, requiring a constant adjustment of oil burners, dampers and auxiliary apparatus, and while the magnitude of these variations was small yet the variation in load was such as to cause a loss in economy as compared with an absolutely uniform load.

56 The economy obtained at the various uniform loads is plotted in the curve shown in Fig. 3. The economy at various fractional loads as compared with the economy at the rated load is also shown in Fig. 3.

57 Inasmuch as the absolute economy at the various uniform loads is probably greater than reported, for reasons pointed out above, and as the correct economy at the uniform load tests would probably be a small but uniform percentage greater throughout all of such tests, it is fair to assume that the relative economies as shown are reasonably accurate, and for this reason the writer considers the curve showing the relationship of economies at various loads to be of greater value than the results of individual tests. There has also been plotted the points corresponding to three in lieu of five boilers, for 2000-kw. load, for both of the above curves, as well as the economy obtained for the official test for the equivalent average load.

58 In studying these curves it should be borne in mind that the results given indicate the entire falling off in fractional load efficiency for the complete plant, being the multiplied effect due to various losses, such as electrical efficiency of generator, mechanical efficiency of engine, indicated economy of engine, auxiliary performance, and the greater effect at light load of dead load losses, such as heat radiation from exposed piping and other radiating surfaces, and the loss in efficiency of boilers owing to operation at fractional loads. As the total effect of all these losses is relatively small, it is in itself evidence of the high maintained steam economy of the prime mover at very light loads. The combined efficiencies of engine and generator are shown in Table 1 and as this falling off in efficiency at light loads is quite apparent, it offers additional evidence as to the maintained indicated economy of engine at fractional loads.

59 A falling off in economy would be expected for the complete plant test of all three units as compared with the official test of No. 2 unit for many reasons; among these are the excess of head imposed on circulating pump on account of pipe line, for reasons explained above; the deficiency in quantity of circulating water, causing a severe loss of vacuum at intervals, and at the completion of test the operation of the plant non-condensing; the use of energy from the plant to operate a 50-h.p. motor driving a 12 by 12 in. duplex vacuum pump, necessary to remove air leakage in suction line; the use of energy in pumping water from the plant for operating motor-driven oil pumps in pumping oil from main oil tanks to auxiliary oil tanks; the use of energy from plant in pumping well water from the purchaser's wells to the plant; the energy used for pumping out sumps; the energy used for lighting various surrounding buildings and wharf; the necessity for hand control of firing the boilers of No. 1 and No. 3 units, the automatic system being in use on No. 2 unit only. Con-

sidering all of these conditions, the economy for the complete plant is very favorable indeed.

60 As having a bearing on the economy of the plant, the writer would explain that to conform to the contract load requirements during test, which are designated as the maximum requirements of the plant during commercial operation, and further to economize in the first cost of plant, the main engines were initially selected to have a normal rated capacity of 4000 kw. in lieu of 5000 kw., the rated capacity of generators. The writer hoped thereby to obtain sufficiently increased economy at the lighter loads to offset the possible loss in economy due to overload. It has been shown that these 4000-kw. engines are capable of carrying overloads in excess of 7000 kw. It appears, however, that it would have been a decided improvement had larger engines been selected, conforming more nearly to the rating of the generators. Due to the use of superheated steam, high fractional load economy would have been insured, the overload capacity still further increased, and superior economy maintained at loads considerably above rating.

61 It is regrettable that the circulating pumps were omitted from the official and uniform load tests. Had the circulating water system conformed to the contract stipulations, or had circulating water conditions existed as in many eastern central stations, where parallel suction and discharge conduits are run directly underneath the circulating water pump, the total head on the system would have been very small indeed. As stated above, the contract stipulation for loss of head in the piping was 5 ft. Had it been possible to conduct the official 15-day test with the circulating pumps, operating under this head, it is safe to say that the additional fuel consumption to operate the circulating pumps would have been scarcely noticeable. The circulating pump engines are of compound non-condensing type and designed for use with superheated steam, insuring a low steam consumption. Inasmuch as the boiler feed water temperature was at all times considerably under 200 deg. fahr., and as practically no exhaust steam was wasted at any period of operation, it is safe to state that all the heat in the exhaust steam from the circulating pump engines would have been returned to the system; compensation for greater steam consumption manifesting itself in a higher feed water temperature.

62 Having no bearing on the economy of the unit, but as a matter of particular note, the writer would state that following the acceptance test on the plant, the engine for the No. 1 unit was operated under

various conditions to determine the possibilities in case of an emergency. It was shown that the unit could be operated on either one, two or three as well as four cylinders. The running conditions were smooth, and the successful parallel operation of the alternators was easily accomplished under all conditions. It was further possible temporarily to overload the cylinders in use during these special trials, to such an extent as to produce in the neighborhood of 1800 kw. per cylinder. The time required to remove a connecting rod in case of an emergency was found to be between one and two hours. It was also found possible to operate the entire engine on one condenser, as well as both condensers, causing but a slight reduction in vacuum, the second condenser being available for repairs, etc. In case of difficulty through sea weed in the circulating water the engines were often run for a long time non-condensing.

63 The writer believes that this type and arrangement of prime mover offers features of flexibility that merit very careful consideration, and enables the maintained operation of steam power plants, under both normal and emergency conditions, with a far less percentage of reserve capacity than is permissible with any other type of prime mover. The Redondo Plant is today operating without any reserve capacity whatsoever, and since the completion of the work it has regularly operated under all conditions of service desired by the management.

64 As the maintained economy of any type of prime mover is of great importance, the writer is able to state that the station records so far fail to indicate any considerable falling off in economy as compared with test conditions. On the other hand, it is a matter of record that a complete plant economy has been attained, since operation of plant by purchaser, exceeding that herein reported. During the dry season this year, and by reason of the large overload capacity afforded by these engines, the management has been compelled, and has been able to operate this station under continuous load far in excess of that for which it was designed. The engines have been operated for long periods at over-loads of 75 per cent above the rated loads, and during such periods only a slight falling off in economy has resulted, which was less than would be expected on account of the special way in which the engines were rated.

65 No. 1 and No. 3 units are now being equipped with an automatic system of firing similar to that on No. 2 and it is expected that the station economy will be materially improved thereby.

COMPARISON OF ECONOMY WITH COAL BURNING PLANTS

66 For the purpose of comparing the economies herein reported with those obtained in modern coal burning stations, the writer has given Table 1 the fuel consumption reduced to the corresponding B.t.u. per kw-hr.; these figures are possibly subject to slight error however by reason of oil corrections already explained.

COMPARISON WITH TURBINE PERFORMANCE

67 To the writer's knowledge, the best turbine performance on the Pacific Coast based on oil fuel is considerably under that found for the Redondo Plant.

68 When comparing these results with economies reported for Eastern turbine stations, the writer would call attention to the necessity of making due allowance for the variation in the initial temperature of the circulating water. The average temperature obtainable in the vicinity of Los Angeles is from 60 to 70 deg. fahr., as compared with circulating water temperatures in the neighborhood of the freezing point obtainable at times in many Eastern stations, this lower temperature enabling the attainment of a considerably improved vacuum. Inasmuch as the steam turbine gains largely in apparent economy due to increase in vacuum, similar reductions should be made in the stated economy of such plants, and correspondingly in the reported fuel consumption.

69 The writer would take this opportunity to urge on the Society the establishment of a standard of performance for comparison of fuel economies of complete plants, to be based on the Rankine cycle, the upper limit being the maximum boiler temperature and pressure, the lower limit for all plants operating condensing, being the initial temperature of circulating water, in lieu of the temperature of engine exhaust. This would be an extension of the standard of performance established by the British Institute of Civil Engineers for the comparison of steam engine performance, in which the Rankine cycle is accepted as ideal, the upper limit being based on the throttle steam temperature and pressure, the lower limit on the temperature corresponding to the engine exhaust.

CONCLUSION

70 The writer believes that the performance herein indicated is noteworthy, and warrants on the part of designing engineers a far more careful and thorough investigation as to the capabilities of steam engines, when used as prime movers, than is manifest in the practice of late years.

INTERCHANGEABLE INVOLUTE GEAR TOOTH SYSTEMS

By Ralph E. Flanders, New York
Non-Member

The standard form of gear tooth for cut gearing has a pressure angle of 14½ degrees and an addendum (in a one diametral pitch gear) of one inch or 1 ÷ P. The pressure angle of an involute gear is the angle that the line of action makes with the common tangent to the pitch circles of two gears in mesh; or it may be stated as the angle made by the side of a rack tooth of a given system, with the perpendicular to the pitch line (see a in Fig. 4). In any system of involute gearing the pressure angle remains the same for all gears in the series. The addendum is the height of the tooth above the pitch line. As this also remains the same for all the gears in a series, a standard system of interchangeable involute gearing may be defined by giving the pressure angle and addendum.

2 While a pressure angle of 14½ degrees, and an addendum of one inch for a one diametral pitch tooth was formerly used almost universally for cut gearing, there have always been occasional departures from this standard. Of late years these departures have become more and more numerous. The pressure angles that have been used have run as high as 28 degrees in some cases, and the addendum has been made as small as 0.6 of the standard height. As this tendency to use shorter addendums or greater pressure angles appears to show no signs of abating, and has resulted in a considerable diversity of forms of gearing in different lines of work, it has seemed worth while to investigate the whole question, to see if these variations are justified; and if they are, whether their justification lies in peculiar conditions, or in a lack of fitness of the standard system for any large part of the work for which it has hitherto been used.

To be presented at the New York Meeting (December 1908) of The American Society of Mechanical Engineers. All papers are subject to revision.

3 As far as possible, the effect of varying the pressure angle and addendum on the various working qualities of gearing in here shown in diagrammatic form, so that the eye can readily grasp the effect of the variations in every case. The method of calculating the various diagrams is described in an appendix,[1] so as to have the body of the paper as free as possible from mathematics, leaving nothing to consider except the results obtained. To any one disposed to question the methods followed in the mathematical investigation the appendix will give full information on this matter. For the sake of simplicity, in all the calculations the teeth are supposed to be of 1-diametral pitch. The pressure angles investigated range from $14\frac{1}{2}$ to $27\frac{1}{2}$ deg. Three heights of addendum are considered: 1.0, 0.8 and 0.6 in. respectively, for 1-diametral pitch. These limits are thought sufficient for the purposes of this paper. It should further be stated that in all calculations where it is necessary to assume the number of teeth in the smallest gear of a series, that number is taken to be 12. the same as in the standard system. The following reference letters are used:

S = height of addendum.
a = pressure angle.
N = number of teeth.
N_{12} = number of teeth in 12-tooth gear, etc.
n = number of teeth in continuous action.

EFFECT OF VARYING THE PRESSURE ANGLE AND ADDENDUM

4 The effect of varying the addendum and pressure angle is investigated with reference to the following practical considerations: interference; number of teeth in continuous action; side pressure on journals; strength; efficiency; durability; permanence of form; quietness and smoothness of action; suitability for practical cutting processes; and miscellaneous practical considerations.

5 Fig. 1 illustrates the effect of changing the addendum and the pressure angle on the question of interference. The lines rising toward the right in the diagram indicate the largest number of teeth in the gear in any given system of interchangeable gearing, which will mesh with a 12-tooth pinion without correction for interference with the flanks of the teeth of the latter. Thus, in a system of interchangeable gears in which $a = 17\frac{1}{2}$ deg. and $S = 0.6$, the dia-

[1] On file in the library of the Society.

gram shows that all gears having more than 50 teeth must be corrected to avoid this interference with the 12-tooth pinion. The lines rising toward the left indicate the minimum number of teeth possible in the smallest pinion in an interchangeable series, to avoid entirely the phenomenon of interference. Thus with the standard form in which $a = 14\frac{1}{2}$ deg. and $S = 1.0$, if 32 be taken as the number of teeth in the smallest gear of the series, instead of 12, it will not be necessary to correct the rack or any other member of the system for interference.

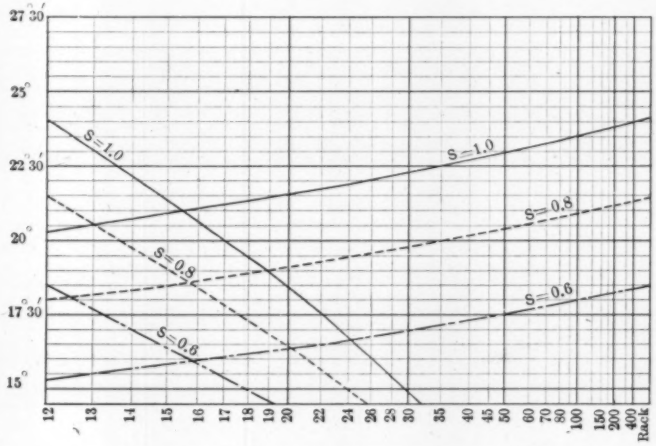


FIG. 1 EFFECT ON INTERFERENCE OF CHANGES IN THE ADDENDUM AND PRESSURE ANGLE

6 The following formulae were used in calculating this diagram. For the maximum number of teeth possible without interference with the 12-tooth pinion,

$$N = \frac{36 \sin^2 a - S^2}{S - 6 \sin^2 a} \tag{1}$$

For the minimum number of teeth possible without interference with an uncorrected rack,

$$N = \frac{2 S}{\sin^2 a} \tag{2}$$

7 It will be seen from the diagram that with the standard form of gearing, in which $a = 14\frac{1}{2}$ degrees and $S = 1.0$, interference occurs to such an extent that the correction of the face of the tooth has to be carried clear down to the smallest gear in the series, it being impossible for two uncorrected 12-tooth pinions to mesh without interference. This condition is shown in Fig. 2. The contact between the two gears, running in the direction indicated and with gear A as driver, takes place along the line CD, being determined by the points of tangency of this "line of action," as it is called, to the "base of circles" of the two pinions. That part of the face of the teeth of gear A (see shaded portion at the right) which lies outside of the interference circle passing through point F, extends beyond any possible contact with the mating pinion, and so is useless for conjugate action in its uncorrected form. Not only is it useless but it is positively harmful as well.

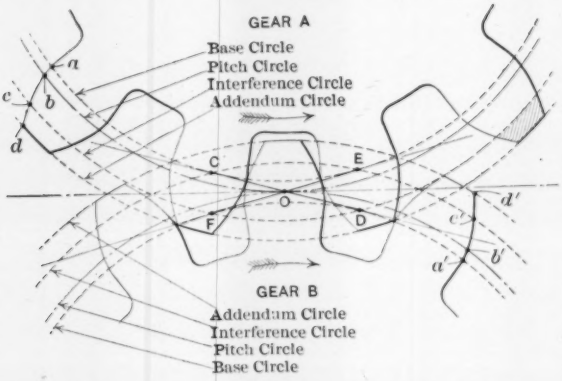


FIG. 2 EXTENT OF INTERFERENCE BETWEEN TWO 12-TOOTH STANDARD PINIONS

8 Fig. 3 shows, in diagrammatic form, the positions of the teeth of the two gears in Fig. 2 at the beginning and end of their action; contact begins at C and ends at D. As the gears continue to revolve, tooth face *aa* of gear A will interfere with tooth flank *bb* near the base circle, making proper meshing of the teeth impossible, no matter what the form given to the flanks of the teeth below the base circle. This phenomenon of interference is discussed in all treatises on gearing.

9 Everything outside the interference line then in Fig. 2 must be corrected for interference. The amount of this correction increases with the number of teeth in the gear, reaching its maximum in the rack, as shown in Fig. 4. As will be explained later, the nature of the

correction depends on the form arbitrarily given to the flanks of the teeth of the smallest pinion below the base circle.

10 . The next thing to consider is the number of teeth in continuous action. This is at the minimum when the two gears having the smallest number of teeth allowed by the system are in mesh with each other and it is at its maximum in the case of the engagement of two racks.

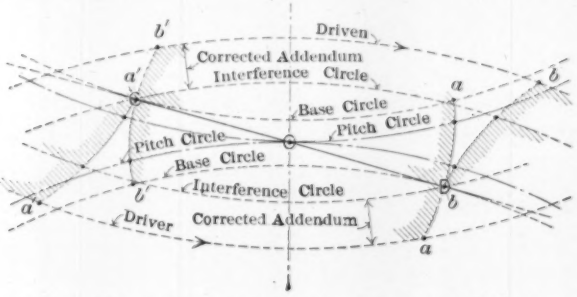


FIG. 3. LIMITS OF TRUE INVOLUTE ACTION WITH TWO 12-TOOTH STANDARD PINIONS IN MESH

A practical problem presented is that of finding the pressure angle and addendum that will give the required minimum action for the two smallest gears selected for the series. The diagram in Fig. 5 offers a practical solution of this problem. Suppose, for instance, it is required to find the minimum pressure angle that will allow 1.1

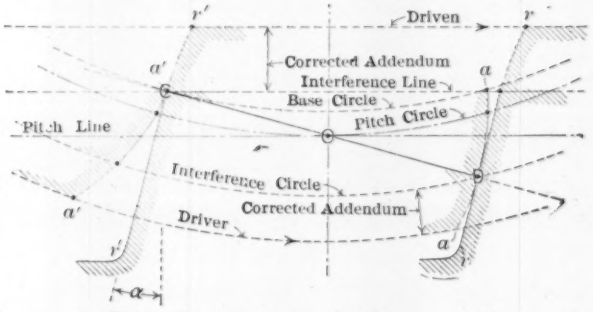


FIG. 4 LIMITS OF TRUE INVOLUTE ACTION WITH A 12-TOOTH PINION AND RACK (STANDARD) IN MESH

teeth to be in continuous action in a pair of 16-tooth gears, when the addendum is 0.8. (By saying that 1.1 teeth are in continuous action, we mean that the line of action is 1.1 times as long as is necessary for continuous action. At times, of course, two pairs of teeth will be engaged, while but one pair will be in action the remain-

der of the time.) The diagram shows that the pressure angle must not be more than 26½ deg. to make this possible.

11 It will be noticed that all the lines running toward the lower left-hand corner, for different values of S but the same values of n, merge with the line marked only for the value of n, the latter line extending downward toward the right. In reading a line for the given value of n and S it is to be traced downward to the left, to its junction with the full line for its value of n, which is to be traced downward. The point of junction of the two lines marks the point where interference ceases for a given value of n and S. For instance, if $n = 1.4$ and $S = 0.8$, the diagram shows no interference in the

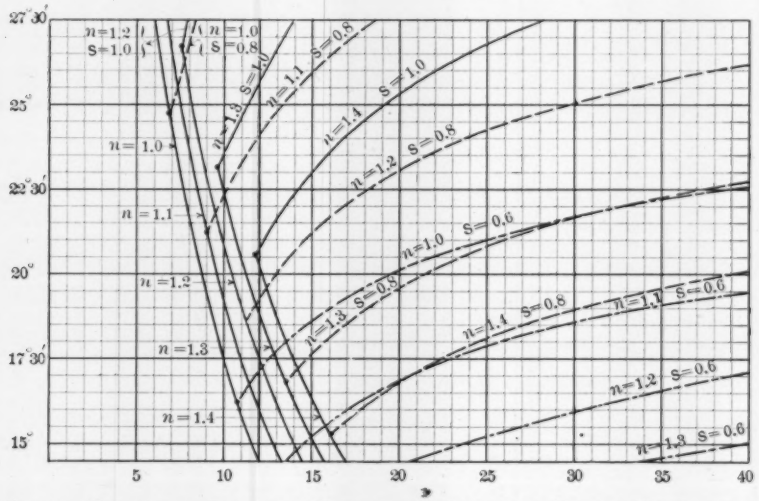


FIG. 5 NUMBER OF TEETH IN CONTINUOUS ACTION, IN MATING PINIONS OF THE SAME SIZE

case of the two smallest, or 16-tooth pinions meshing with each other, and so there is no interference in any gears of this series.

12 This change in the direction of the line indicates that many cases have two solutions. For instance, let it be required to find the proper pressure angle for 1.1 teeth in continuous contact, in a pair of 12-tooth gears, when $S = 0.8$. Following the curve $n = 1.1$, $S = 0.8$, we find that it crosses the 12-tooth ordinate at about 24 deg. 12 min. Following this curve down to its junction with the main curve, $n = 1.1$, we find that this crosses the 12-tooth ordinate again at about 16 deg. Thus there appears to be, as there are, two solutions to the

problem. These two results are shown diagrammatically in Fig. 6 and 7. In the first case, above the interference point, the action is limited by the points E and F, determined by the addendum circles for $S = 0.8$. In the second case, where interference occurs, the action is limited by points C and D, the points of tangency with the base circles. The amount of action with these two different angles is the same, there being 1.1 teeth in continuous contact in each case. In Fig. 6 the teeth are in conjugate action clear out to their points, while in Fig. 7 these points have to be corrected for interference. In Fig. 6, making $S = 1.0$ increases the number of teeth in contact by lengthening the line of action to E′F′. In Fig. 7, it will be seen, a similar increase of addendum makes no change in the number of teeth in contact, as the action is still limited by points C and D.

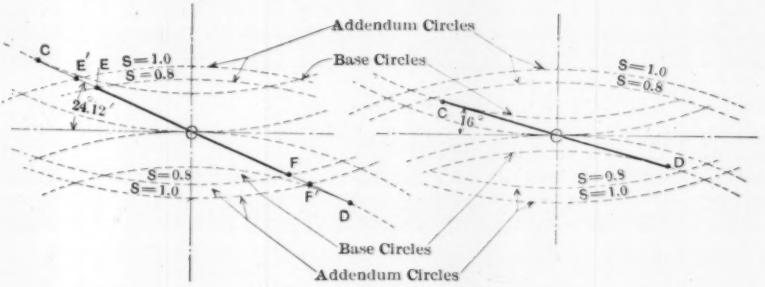


FIG. 6 LIMITATION OF CONJUGATE ACTION BY THE ADDENDUM CIRCLES

FIG. 7 LIMITATION OF CONJUGATE ACTION BY INTERFERENCE

13 The following formula was used for plotting the curves above the interference points in the diagram in Fig. 5:

$$N = \frac{(\frac{n}{2}\pi \cos a)^2 - S}{S - \frac{n}{2}\pi \sin a \times \cos a} \tag{3}$$

14 The following formula was used for plotting the curves below the interference point:

$$N = \pi n \cot a \tag{4}$$

15 Fig. 8 shows the maximum and minimum number of teeth in contact, for any system of interchangeable involute gearing in which the 12-tooth pinion is the smallest. The minimum contact occurs in the case of two 12-tooth pinions in mesh, and the maximum in the impractical case of two racks in mesh. As will be seen, the 14½-deg. stand-

ard series of whatever height of addendum, gives less than continuous action, being about 0.987. It will also be noticed that below the points of interference for the 12-tooth pinion (represented by the junction points of the maximum and minimum lines with the interference line) the contact is constant. Thus, for a system in which $\alpha = 20$ deg. and $S = 1.0$, the amount of contact between any two gears of the series from 12 teeth to a rack, is constant at about 1.40, while for a system in which $\alpha = 22$. deg. 30 min. and $S = 1.0$, the amount of contact varies between 1.36 for a minimum and 1.58 for a maximum. Fig. 3 and 4 show this condition, which indicates that if in any series there is interference in the case of the two pinions having the

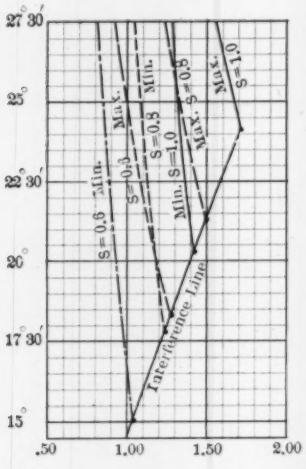


FIG. 8 MAXIMUM AND MINIMUM NUMBER OF TEETH IN CONTINUOUS ACTION

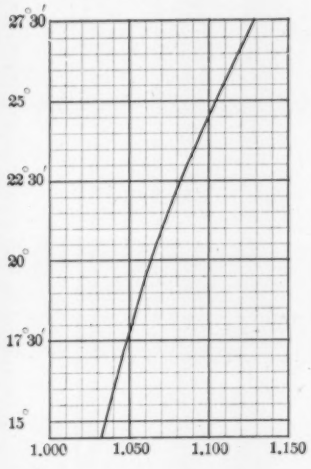


FIG. 9 SIDE PRESSURE ON BEARINGS

smallest number of teeth allowed by the series, the amount of action obtained in that case is constant for any other case throughout the whole series, up to that of two racks meshing with each other. As far as the author knows, this condition has never before been noticed.

16 In Fig. 3, which shows two minimum pinions of a series meshing with each other, the action is limited to the line CD. In Fig. 4, which shows the minimum pinion meshing with a rack, the action is still limited to the same line CD. It cannot extend beyond C in one direction, because it cannot pass the point of tangency in the base circle of the pinion. It cannot pass beyond D in the other direction, because the points of the pinion teeth are corrected beyond the

interference circle, losing their true involute form. In the case of any other gear meshing with a rack, the action is limited to C on one end owing to the correction for interference of the points of the rack teeth, and at D on the other end owing to the correction for interference of the points of the gear teeth. And in the case of any two gears, the action is similarly limited to the line CD by the corrections for interference at the points of the teeth. As may be seen, then, a true involute system in which $\alpha = 14\frac{1}{2}$ deg. and S = 18, just fails of continuous conjugate action in the case of any two gears of the series.

17 The formula used for calculating the maximum curves of Fig. 8, above the interference points, is as follows:

$$n = \frac{2\,S}{sin\alpha \times cos\ a \times \pi} \tag{5}$$

18 The formula used for calculating the minimum curves is as follows:

$$n = \frac{2}{\pi\ cos\ a}\left[\sqrt{(6 + S)^2 - (6\ cos\ a)^2} - 6\ sin\ a\right] \tag{6}$$

19 The formula for calculating the interference line, which gives the contact below the interference points, is the same as formula 4 when $N = 12$:

$$n = \frac{12\ tan\ a}{\pi}$$

20 One of the great practical advantages of the involute gear is the possibility of varying the center distance without interfering with true conjugate action. The comparative amount of separation possible for any two proposed systems, without losing continuous action, may be estimated from Fig. 8 by comparing the amounts of continuous action for the two cases. Naturally the one having the greater number of teeth in continuous action will stand more separation than one which has less. This point could be calculated, but by processes so devious that it did not seem worth while to spend the time for it.

21 With increase of the pressure angle there is an increase in the side pressure on the journals. Since the side pressure of the journals is the resultant of the tangential pressure between the mating teeth at the pitch line, and the radial outward thrust due to the augularity of the meshing surfaces at the pitch line, it varies directly with the secant of the angle. The curve shown in Fig. 9 is therefore a secant curve.

22 The rational formula for the strength of gearing is the well known one developed by Mr. Lewis.[1] For diametral pitch measurements, this formula takes the following form:

$$W = \frac{SF_y}{P}$$

in which W = the load transmitted by the teeth in pounds.
S = the safe working stress of the material.
P = the diametral pitch.
F = the face in inches.
y = a factor depending on the form of the teeth.

As the strength varies directly with y, this factor forms the basis of comparison for different forms of teeth.

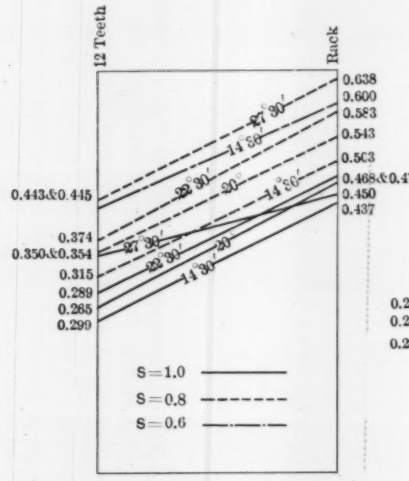


FIG. 10 MAXIMUM AND MINIMUM VALUES OF STRENGTH FACTOR FOR TYPICAL CASES

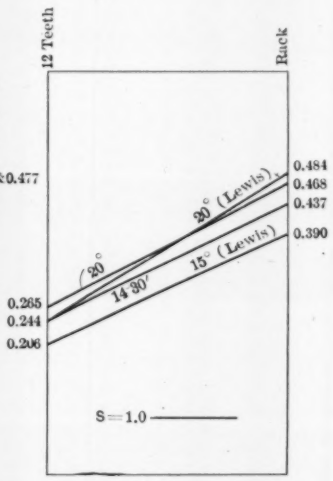


FIG. 11 COMPARISON OF VALUES FOR y HERE OBTAINED, WITH THOSE CALCULATED BY MR. LEWIS

23 Fig. 10 shows the maximum and minimum values of y for certain selected forms of gear teeth. It will be seen that the values of y rise rapidly with the decrease of the addendum, and more slowly with an increase of the pressure angle, which does not affect it so much.

24 This y factor was obtained by the method laid down by Mr. Lewis, with the incorporation of a slight refinement, taking into account the fact that the pressure is applied to the teeth at a point

[1] Kent's Pocket Book, p. 901.

above the pitch line. In order to get the values of y in the case of gears in which there is interference, the form of the flank of the 12-tooth pinion below the base line had to be arbitrarily determined. The proper corrections of the faces of the teeth were obtained by considering the flanks of the 12-tooth pinion to be radial below the base line. This gave to the rack teeth a cycloidal form above the interference line, generated by a rolling circle of half the pitch diameter of the 12-tooth pinion. In order to get the proper form of the fillet on the 12-tooth pinion, the rack tooth was lengthened by an amount equal to the clearance, and the corner of this extended tooth was rounded with a radius equal to $\frac{2}{3}$ of the clearance. The pitch line of this extended tooth was rolled on the pitch line of the 12-tooth pinion, and the fillet thus generated used in obtaining the factor of y. As the shape of the fillet is of great importance in strengthening a gear tooth, it was thought best to determine it rationally as just described, its form being that which would be given it by a generating process such as that of the hobbing machine.

25 Fig. 11 shows that the values obtained for y vary somewhat from those obtained by Mr. Lewis (it will be noted that his values have been multiplied by π to make the formula read for diametral pitch). This variation is probably due to the fact that in this investigation the outlines from which the factors are calculated have their teeth corrected for interference, thus bringing the normal pressure at a greater angle at the points of the teeth; and to the fact that the fillets have been constructed by the method just described.

26 The work lost in friction in the case of two gears meshing with each other is proportionate to the product of the rate of sliding and the normal pressure on the surfaces in contact. This rate of sliding varies directly with the distance of the point of contact from the pitch point 0 (Fig. 3 and 4, etc.) of the gears. The pressure is constant throughout the action, except that when two teeth are in contact it may be considered that the pressure is evenly divided between them. The diagram in Fig. 12 was calculated by a modification of the method devised by an English engineer, Mr. Bruce,[1] the modification being simply that of considering the pressure on two sets of teeth as evenly divided between them, when there are two sets of teeth in continuous action. His method was also corrected to allow for the increase of pressure with the increase of α, and to take into account the time scale for distances on the line of action. The

[1] American Machinist, October 10, 1901

calculations are made for a single case of gearing, an 18-tooth pinion driving a 60-tooth gear. This case was selected as typical.

27 The diagram for durability is shown in Fig. 13. The wear between the teeth of two gears is proportionate to the continued product of the pressure, the rate of sliding, and a third factor which depends on the shape of the surfaces in contact. This factor is greater for sharply curved convex surfaces and less for a large radius curve rubbing on a flat surface, for instance. This being the case, it seemed to the author that a reasonable basis of comparison for different systems of gearing could be made by multiplying the values given

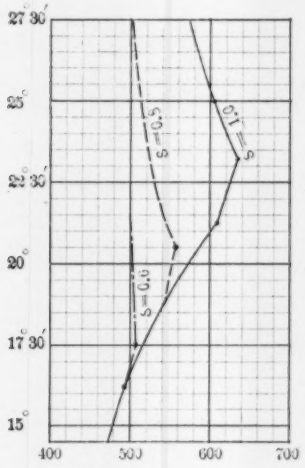


FIG. 12 COMPARATIVE AMOUNT OF LOST WORK

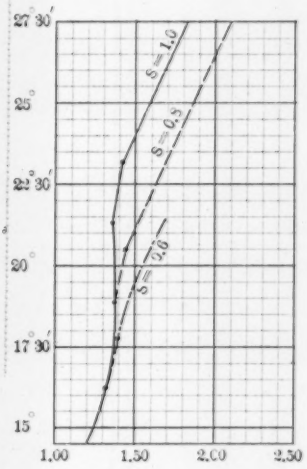


FIG. 13 COMPARATIVE DURABILITY

in the diagram of Fig. 12 by a factor determined from the shapes of the surfaces in contact, and plotting the reciprocal. In determining this factor, the author has followed Mr. C. E. Logue,[1] who obtains it by adding together the sums of the curvatures (that is, the sums of the reciprocals of the instantaneous radii) of the gear teeth at the pitch line. This should give a value which is approximately an average of the conditions existing throughout the entire action. (It should be understood that Fig. 12 and 13 indicate comparative values only. Positive calculations of efficiency or durability are not here contemplated.)

[1] American Machinist, February, 1908

28 The question of permanency of form may be considered aside from that of durability. By durability we mean the lasting quality of the gear, without reference to whether or not it keeps its shape in the wearing process. A gear which has permanency to a high degree might wear out very rapidly, but it would retain nearly its true form throughout the whole of its short life. In Fig. 14 are given a series of diagrams showing the tendency of a correctly formed tooth to wear at different points of the tooth outlines, in the case of a 12-tooth pinion and the rack, and for the same pressure angles and addendums that were selected for the strength diagram of Fig. 10.

29 Besides the various factors for which we have given diagrams, there are a number of important ones of such a nature as to be incalculable. One of these factors is the suitability of a system for use in different methods of tooth cutting. Any form of tooth which involves interference is very unsatisfactory for use in any generating process for instance. The formed cutter process works with equal facility on any form of tooth, except that the increase in the pressure angle gives somewhat more side clearance, leading to a freer cutting action.

30 The vital incalculable factor is smoothness of running. Smoothness of action depends theoretically on perfect conjugate action between the mating teeth. In practice it depends as well on accuracy of cutting tools and accuracy of machine setting. The cutter may be set out of center, or set deeper than required, or have its face ground at an angle considerably away from the radial plane required to give the true cutting shape. The standard form of gear tooth, developed by the Brown & Sharpe Manufacturing Company, has been improved by long practical experience to a point where it takes care of these practical inaccuracies in a very satisfactory way. This is possible, it appears to the writer on account of the considerable portion of the outline which is indeterminate in form, namely, that part of the outline outside of the interference circle and inside of the base circle (see Fig. 2). These can be given any shape shown by practice to be proper for leading the teeth gently into action with each other at high speeds even with inaccurate cutting. Of course this shaping is entirely founded on experience.

31 It might be mentioned that the inaccuracies just mentioned as occurring in practical gear cutting are those chiefly met with in the formed cutter method. The inaccuracies of the generating processes are mostly due to difficulties in forming the cutters, and in taking care of the strains in the machine without deflecting the mechanism.

Looked at from this standpoint, it may be that the standard form is best for formed cutter work, while larger angles and shorter addendums are more suitable for generating processes.

RELATIVE IMPORTANCE OF THE VARIOUS CONSIDERATIONS

32 As to the importance of the various considerations mentioned above as being affected by changes of the pressure angle and addendum, it may be said that the difficulty due to interference (apart from its effect on strength, efficiency, etc., which is considered under those heads) is that of making the shape of the gear indeterminate. The design of the standard gear is empirical. While the writer has followed the plan of making the flanks of the 12-tooth pinion radial, and generating the fillet by means of an extended and rounded rack tooth, produced by such a radial flank, he does not know that this is the form of the standard tooth. The exact form is known only to the makers of standard cutters and is not public knowledge. It will be seen that this matter of interference makes of the standard system practically a short tooth system so far as the theoretical bearing is concerned. This is plainly shown in Fig. 14, for Case 1, which is the standard involute gear tooth. The bearing extends over but a very small part of the tooth face and flanks. The bearing can, of course, be carried clear to the points of the teeth by making the non-involute parts of the teeth on some other conjugate system, as was done, for instance, by the author in par. 24 by the cycloidal corrections. This would give more action than Fig. 8, but since only a part of it is involute, the excess of action would be lost as soon as the center distance is changed. A conjugant correction of this kind is made by the manufacturers of standard cutters.

33 The number of teeth in continuous action, shown in the diagram of Fig. 8, is of importance principally in relation to smoothness of action. It is not generally considered possible to count on distributting the load evenly between two teeth in calculating the strength of gearing, though it will be noted that where n is slightly more than unity, two teeth are in bearing at the beginning and end of the action, when the fiber stress in the teeth is at the greatest. In the matter of smoothness of action, it is necessary to have one pair of teeth take up the load before the previous pair drop it. Increased smoothness of action, due to smoothing out irregularities in cutting, could be effected by having two, three or more sets of teeth in action at one time, but this is impracticable with interchangeable involute gearing. A

considerable length of contact is also of advantage in permitting a considerable variation in the center distance, without loss of continuity of action, as previously explained. In this respect the increased pressure angles will be seen to have considerable advantage over the standard form.

34 The side pressure, shown in Fig. 9, is an almost negligible factor. As may be seen, the increase for even so great an angle as 27½ deg. is only 9 per cent above that given by the standard form. At

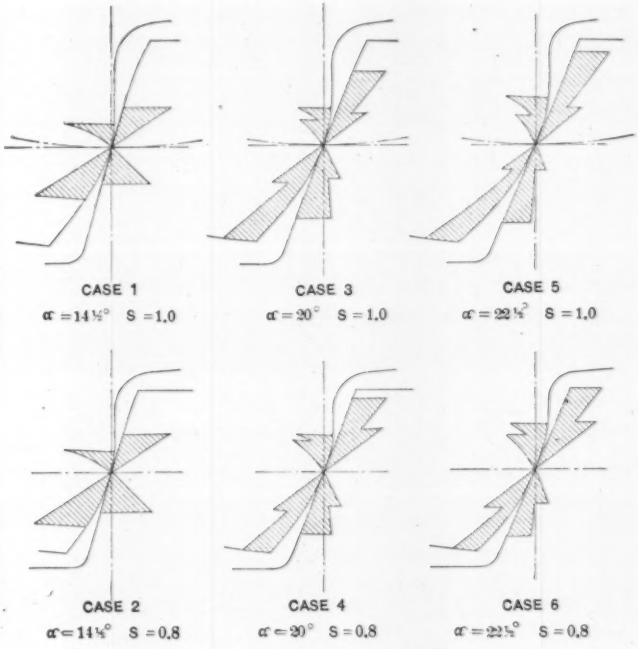


FIG. 14 PERMANENCY OF FORM

22½ deg. it is 3⅓ per cent above. As the shaft bearings can take care of this slight increase of side pressure in a perfectly normal and satisfactory way, the question scarcely enters into consideration.

35 The matter of strength is an important one. If a stronger form of tooth can be used, many mechanisms can be gotten into a much smaller space than is otherwise possible. A case in point is the design of geared speed and feed mechanism for machine tools, in which the space problem is usually serious. By decreasing the

face of the gear to correspond with an increase in the strength factor y, the problem of the designer would be greatly simplified. Instead of reducing the width of the gear by using a stronger form of tooth, advantage can be taken of the stronger form to increase the number of teeth and make them of finer pitch. This gives smoother action, owing to the small inaccuracies in the fine teeth. Smoother action at high speed also means increased strength, owing to the lessening of stresses due to impact. It is becoming recognized that impact at high speeds is greatly affected by the accuracy of the surfaces in contact; and even where the strength is sufficient the matter of noise in high speed gearing is of great commercial importance. People would not buy noisy gearing if they could help it, even if it fulfilled their requirements in every other particular.

36 The question of lost work (plotted in the diagram of Fig. 13) is not a matter of serious moment. The efficiency of well-made spur gearing is so high[1] that the slight variations indicated by the diagram are of small practical importance. This diagram of course takes no account of the increase in lost work due to increase in journal pressure, which is an inconsiderable factor with well made bearings.

37 Durability is of much more importance. Gears have been almost invariably designed hitherto for strength rather than durability, but it is becoming recognized that in many cases the principal factor in gear design is the wear of the teeth. If then the methods by which the table of Fig. 13 is calculated are rational, the results there shown are of considerable importance.

38 Permanency of form is another important factor. Gears which the writer has seen worn out in hard service, show very plainly the severe concentrated wear inside the pitch line indicated in Case 1 of Fig. 14. The advantageous effect of an increased pressure angle on permanency is worthy of consideration.

39 The fitness of a tooth form for various cutting processes is of importance in determining the future of the apparently attractive generating method of cutting gear teeth. One form of generating process has come into extensive use in spite of the handicap of the $14\frac{1}{2}$-deg. angle, with the various corrections for interference required. There is reason to believe, however, that this process works better with forms of teeth in which the angle is increased or the addendum decreased. In the matter of the hobbing process, which has developed so rapidly in the past two or three years, the author must say that

[1] Mr. Lewis' experiments, Kent's Pocket Book, p. 899

he has seen comparatively little gearing cut by this process that seemed entirely satisfactory. This opinion will doubtless be disputed by all the makers and most of the users of these machines; but the difficulties of the process are recognized by everyone concerned, and many of them would be removed by a change in the form of tooth. Were the change advisable for other reasons, this fitness for generation would be an added advantage, though it would scarcely be worth changing on this score alone. So far as cutting teeth by the formed cutter process is concerned, an increase in the pressure angle is favorable, as more side clearance is obtained for the cutter; while decreasing the addendum gives a less volume to cut out, and a consequent cheaper production of gears. It might also be mentioned in this connection that the decrease in face or pitch that could be obtained by the use of stronger forms of teeth would also result in cheapening the gear, owing to the fact that less metal would have to be removed.

TYPICAL DEPARTURES FROM THE STANDARD FORMS

40 Among the various departures from standard practice which have been made from time to time, and especially in recent years, the writer will mention a few typical cases. The first is that of William Sellers & Company who have for many years used a form of interchangeable gearing in which $S = 0.942$ and $\alpha = 20$ deg. This form was determined by Mr. Wilfred Lewis, and is said to have given continuous satisfaction. A form of gear in use in the practice of the C. W. Hunt Co. retains $14\frac{1}{2}$-deg. pressure angle, but uses a shorter addendum in which $S = 0.785$. While the Sellers Standard was adopted principally for the avoiding of interference and for the increase of strength obtained, the Hunt standard is based on the increase of strength alone, which is, as shown by the diagram in Fig. 10, more effectively obtained by reducing the addendum than by increasing the pressure angle. Mr. Hunt states that this gearing has given unexceptional results in smoothness of action, as well as being stronger. What is practically the Sellers form of tooth, except that $S = 1.0$ has recently come into limited use in machine tool design, particularly in the care of the geared drive for heavy duty spindles.

41 In rolling mill work it has been the practice for many years to use pressure angles as high as 28 deg. which angle has been employed the author understands, by the Carnegie Steel Company. $22\frac{1}{2}$ deg. is a very common pressure angle. The addendum varies, being

sometimes of standard height and sometimes considerably less. A typical example is that of the Wellman-Seaver-Morgan Engineering Company of Cleveland, O. Through the courtesy of Mr. Stratton, Engineer of Construction, the writer has received blue-prints of the lay-out for the gears of a 22-in. blooming mill, in which the pressure angle is 20 deg. and the addendum 0.785. Even shorter forms of teeth have been used by this company in their gearing for their automatic water hoist, in which the pressure angle is 20 deg. and the addendum about 0.6. In rolling mill work in general, of course the prime consideration is strength, and this is undeniably obtained by such departures from standard form.

42 A form of tooth that has found wide use, particularly in automobile work, is the so-called "stub tooth," developed by the Fellows Gear Shaper Company. In this the pressure angle is 20 deg., while the addendum varies from 0.7 to 0.8, depending upon the pitch of the gear. While the particular way in which this variation of the addendum is taken care of is very convenient for the operator of this builder's machine, it seems to the writer a mistake not to have made it constant for all pitches. This is aside from the question of its general suitability however.

43 In reply to letters sent out to automobile builders (all the members of the American League of Automobile Manufacturers), the writer received thirteen replies, in which seven stated that they were using the "stub tooth" while four were using the standard form. Of those using the special gearing five stated that they had proved in service their ability to run as smoothly as the standard form, and of these, two had obtained even better results from them in this respect. Smoothness of action was not mentioned in the inquiry, so that the information on this point was purely gratuitous.

COMPARISION OF TYPICAL INVOLUTE TOOTH SYSTEMS

44 It is proposed to give, in connection with this discussion of the effect of the variation of pressure angle and addendum, a comparison, by the diagrams herewith presented, of various typical forms of gearing in relation to the practical points just discussed. For the purposes of this discussion the following forms are considered:

Case 1: $\alpha = 14\frac{1}{2}$ deg., $S = 1.0$.
Case 2: $\alpha = 14\frac{1}{2}$ deg., $S = 0.8$.
Case 3: $\alpha = 20$ deg., $S = 1.0$.
Case 4: $\alpha = 20$ deg., $S = 0.8$.
Case 5: $\alpha = 22\frac{1}{2}$ deg., $S = 1.0$.
Case 6: $\alpha = 22\frac{1}{2}$ deg., $S = 0.8$.

45 Case 1 is the standard form. Case 2 is practically the Hunt system; 3 is the Sellers system; and 4 is practically the "stub tooth" system. The last two columns show the effect on the Sellers and "stub tooth" systems respectively, of increasing the pressure angle from 20 to 22½ degrees. The table, in which these various forms with their corresponding advantages and defects are tabulated, suggests that by a change in the present standard of gearing, marked advantages could be obtained on the following scores: avoidance of interference, giving a greater number of teeth in true involute action, and

COMPARISON OF SELECTED EXAMPLES OF INVOLUTE GEAR TOOTH SYSTEMS

Point of comparison	Case 1 B & S Standard	Case 2 Hunt Standard (approximate)	Case 3 Sellers Standard	Case 4 "Stub tooth" (approximate)	Case 5	Cas 6
Smallest pinion in series, to avoid interference	32	26	17	14	14	11
Maximum number of teeth without correction for interference with 12-tooth gear	None	None	None	36	35	Rack
Maximum and minimum number of teeth in continuous contact	0.98	0.98	1.38	1.38 1.18	1.58 1.36	1.44 1.13
Proportion of side pressure on bearing to tangential pressure	1.033	1.033	1.064	1.064	1.082	1.082
Strength factor of rack	0.437	0.503	0.468	0.543	0.477	0.583
Strength factor of 12-tooth gear	0.244	0.315	0.265	0.354	0.289	0.374
Comparative loss of work from friction	475	475	572	552	628	532
Comparative durability	1.21	1.21	1.39	1.44	1.40	1.63
Permanency	See Fig.	14				

For incalulable factors, see text.

permitting more separation with continuous contact; increase of strength; increase of durability; increase in permanency of shape; and increase in fitness for generation and cutting by formed cutters. On the other hand, such changes means an increase in side pressure, comparatively little change in efficiency, and an incalculable and unknown change in smoothness of action. Side pressure and efficiency (as has been shown) are of comparatively minor importance. Smoothness of action is of prime importance. The results of a change,

in this respect can only be arrived at by learning the experience of the users of various forms of gearing, and the author hopes for a full discussion in order that every known fact relating to smoothness of action may be brought to bear on this question.

46 Whatever the condition relative to high speed work, however, it seems to the author that the preceding discussion points clearly to the wisdom of an alternative gear tooth standard of shorter addendum and increased pressure angle; for such work as heavy mill gearing and slow speed gearing in general in which smoothness of action is not a prime requirement. Increased strength and permanence of form would appeal particularly to engineers whose work permits them to design machinery by rational rather than by empirical methods.

47 Aside from the specific cases mentioned, correspondence and conversation with makers of cut gearing bears testimony to a growing demand for, and use of, a stronger form of tooth. The adoption of a standard for this work would give all the advantages which come from standardizing: the preparation of tables of dimensions and of strength; the devising of odontographic tables for the fillet at the root of the tooth, and for the correction—if any is needed; and the reduction in the stock of form took, necessary for doing a general line of gear cutting work.

48 In any event, a standard of gearing of known form, which can be laid out by any engineer who desires to employ it would be an advantage, if such a consummation is possible without sacrificing any of the good qualities which we are accustomed to expect from the present standard.

SPUR GEARING ON HEAVY RAILWAY MOTOR EQUIPMENTS

By Norman Litchfield, New York
Non-Member

In the operation of ordinary street railway motors the gearing is not a serious factor on account of the low horse power required and it is not until the equipment becomes similar in proportions to that of a steam railroad with congested traffic that its importance begins to be felt. With the installation of electric train service on the Manhattan Elevated Railway in 1901, however, the gearing question forced itself on the attention of the engineers through the large number of failures immediately occurring. On the New York Subway equipments the breakages were still more numerous, for these trains are more powerful than any heretofore used, an eight-car subway express train having motors aggregating 2000 h.p., equivalent to a locomotive of about the same power as the new electric locomotives of the New York Central Lines, but differing from the latter in that all the power is transmitted by gears, while the Central's locomotives are gearless.

2 It is evident that the greatest work done by the gearing on any locomotive is during the period of acceleration, and that for a given mileage the total amount of work done will depend upon the rate of acceleration, weight accelerated per gear, and total number of such accelerations or starts. During the evening rush hours in the New York Subway, the total load per gear is about 35 tons, and this weight has to be accelerated at the rate of 1.25 miles per hour per second every third of a mile, that being about the average distance between stations.

3 How serious a matter the breakage of one gear in the rush hour becomes will at once be realized when we state that at one of the many stub end terminals of the system 37 trains in and out per hour are handled during the period of maximum traffic. For the purpose of comparison the train movement at the Grand Central Station of

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the New York Central Railroad may be cited, the maximum number of trains there being 35 per hour, and this is, we believe, considered rather extraordinary congestion. One such breakage in the evening rush hour therefore means an inevitable delay to service of half an hour or more and the disruption of the train schedule for the rest of the day, thus entailing great discomfort to the traveling public.

4 Both the elevated and subway trains of New York are operated by the Interborough Rapid Transit Company and there is no doubt that the conditions under which their trains operate are unique, further evidence being furnished by the failure of various apparatus other than gears which had proved entirely successful on other roads. We believe therefore that the data obtained by this company may be taken as measures of the limitation of power transmission by gearing in railway service.

5 Extended studies which we have recently made tend to show that the question of strength has not heretofore received the attention it merited, but that the line of progress followed most sedulously until very recently has been that of increasing the resistance to wear. That this is a highly desirable object cannot be denied, but our experience with the Interborough equipments has shown the most pressing need to be to overcome absolute failure.

6 It is this phase of the question (absolute failure) which we have felt would be of interest to the Society, and so present in this paper the data which we have brought together on the subject.

7 As is probably familiar to all, the electrical operation of the Manhattan Elevated was commenced some three years previous to the opening of the New York subway, and our earlier experience was therefore with the equipment for the former. The initial gearing installation on the 125 h.p. motors consisted of wrought steel pinions and solid cast steel gears of 3-diametral pitch, this pitch being adopted on account of the economical current consumption thereby obtained. The pinions at once began to fail at the rate of about 15 per month. The failures continuing, it was decided to withdraw all of the gearing then in service and replace it with $2\frac{1}{2}$ pitch on account of the greater tooth section thereby obtained, although, as before stated, this meant some loss in economy of current consumption. This change practically ended the failure of the pinions, but not entirely of the cast gears, and it was decided that greater reliability could be obtained by adopting a composite type of gear consisting of a cast steel centre on which a wrought steel rim was shrunk. This combination of a wrought steel pinion and wrought rim gear of $2\frac{1}{2}$ pitch has proved generally

satisfactory for the elevated service, and the improvement to be looked for is therefore in the line of greater wearing life.

8 On the Subway Division the motors are of 200 h.p. each and the original gearing equipment consisted of solid cast steel gears with wrought steel pinions, diametral pitch $2\frac{1}{2}$, and teeth of the Brown and Sharpe standard $14\frac{1}{2}$ deg. involute. As on the Manhattan Division, so on the Subway, it soon became evident that the design was not proper, but in contrast to the Elevated it was the gears which first caused the trouble. The cast steel gears, therefore, were all scrapped and replaced by the wrought rim type. (It should be stated that the idea of these composite gears was induced by the experience of the Interborough Company, although they were first brought out in connection with the proposed electric motor car equipments for the New York Central Lines.)

9 This improvement practically ended the gear breakage, but unfortunately the pinions began to go, the breakages averaging over one a day, and furthermore we find it unsafe to run a pinion the teeth of which measure less than $\frac{3}{16}$ in. at the top. We are therefore compelled to scrap material which should be available for wear.

10 To what then, shall we look as a remedy? Three suggestions have been advanced by the gear manufacturers:

a Diametral pitch less than $2\frac{1}{2}$.

b Steel with elastic limit of 90 000 lb. per sq. in. and over as compared with our present 45 000.

c 20 deg. stub teeth.

11 The first suggestion we are unable to accept on account of some local conditions, but the combination of the other two seems to have possibilities of success and we are now replacing all our gearing with specially treated carbon steel with stub teeth.

12 Let us now consider the loads the teeth have to carry and determine why it is that we have to resort to the use of special designs and material in order to make our gearing stand up.

13 Fig. 1 illustrates an express run in the Subway, giving the power consumption per train of eight cars, five of which are motor cars, each carrying two motors, thus making ten gears per train. On the same diagram curves are plotted showing the fibre stress in a standard $14\frac{1}{2}$ deg. involute and a special 20 deg. involute tooth both worn to our present limit of $\frac{3}{16}$ in. at top of tooth, the tooth outlines being shown in Fig. 2. The force acting at the pitch line of the tooth has been figured from the power consumption curve and the motor

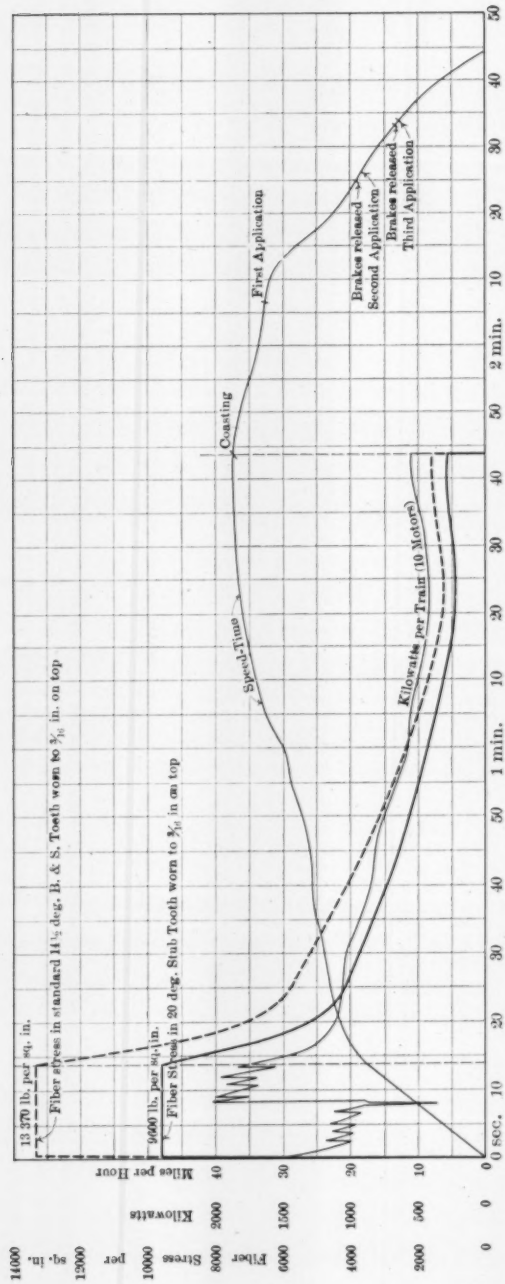

FIG. 1 STRAINS IN TEETH DURING EXPRESS RUN—SUBWAY DIVISION (CALCULATED BY LEWIS METHOD)

characteristics, and the fibre stress in the tooth has then been calculated by the method outlined by Mr. Wilfred Lewis in his paper before the Philadelphia Engineers' Club in 1893, which consists in assuming that all the load is carried by one tooth over its entire face, but at the extreme top, so that the tooth is considered as a beam loaded at one end (the crest) and supported at the other (the base). He states that the force which can be safely transmitted lessens as

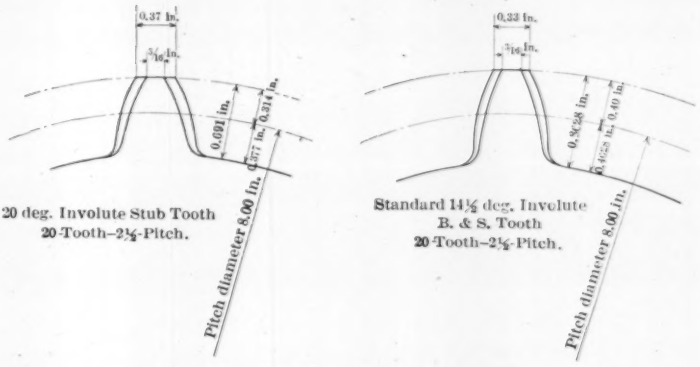

FIG. 2 OUTLINES OF NEW AND WORN TEETH

the speed is increased and gives figures which may be used as safe working stresses at various speeds. These values when plotted give a curve which may be closely represented by the equation

$$S' = S \frac{600}{600 + V}$$

in which

S = safe working stress in pounds per square inch at no velocity.

S' = Safe working stress in pounds per square inch at velocity, feet per minute,

V = velocity of pitch circle in feet per minnte,

14 Referring to the diagram it will be noted that the worst condition occurs just at the commencement of the motor curve, this being the point at which the gearing is transmitting the maximum torque at the greatest speed obtained during the period of acceleration, the maximum torque occurring of course, only during this period. Under the conditions shown the fibre stress at this point is 13 370

lb. per sq. in. in the standard 14½ deg. involute tooth, and at this instant the train is running at 17.3 miles per hour, with a corresponding gear speed of 1168 ft. per minute. On account of this speed, therefore, we require a material which will permit the use of a safe working stress at no speed,

$$S = 13\ 370 \times \frac{600 + 1168}{600} = 39\ 400 \text{ lb. per sq. in.}$$

The elastic limit of the material we have had heretofore runs about 45 000 lb. per sq. in. and we therefore have a factor of safety of only 1.1, which is obviously low.

15 In these calculations we have considered the elastic limit of the material rather than the ultimate strength, on account of the dynamic character of the load, each pinion tooth receiving nearly 1800 blows per mile. The greater the ratio, therefore, between the fibre stress and the elastic limit of the material (the other physical properties remaining of proper value), the greater the life, and some relief should therefore be obtained by the adoption of the special 20 deg. stub tooth which reduces the fibre stress nearly 30 per cent and increases the minimum factor of safety from 1.1 to nearly 1.6.

16 A much greater relief, however, may be looked for by the use of steel with a high elastic limit, say 90 000 lb. per square inch, which used in conjunction with a design of stub tooth increases our minimum factor of safety to 3.2 as compared with our present 1.1.

17 It will be noted that the foregoing calculations are all dependent upon the use of the Lewis formula, the fundamental assumption of which is that the teeth bear across the whole face. Consideration, however, must be given to the fact that in practice it is not possible always to maintain perfect alignment between gear and pinion on account of the necessarily rough design of the motor and its assembly on the truck. The effect of this disalignment is shown in Fig. 3, which is a photograph of a pinion which broke in service, and in the side view it is seen that the lower half of the teeth have been battered over by the loose piece jamming between the gear and pinion. It is the upper half of the teeth which is cracked, however, showing that these cracks were not caused by the jamming of the loose piece, but by the gradual hammering of the pinion against the gear in service. It will be noticed that points *a*, *b*, *c*, *d*, *e*, gave away successively until at last the loose piece was caught and ripped out, and we find that this is typical of all the failures. The break ordinarily starts at the inner face of the pinion, due to the natural tendency of the arma-

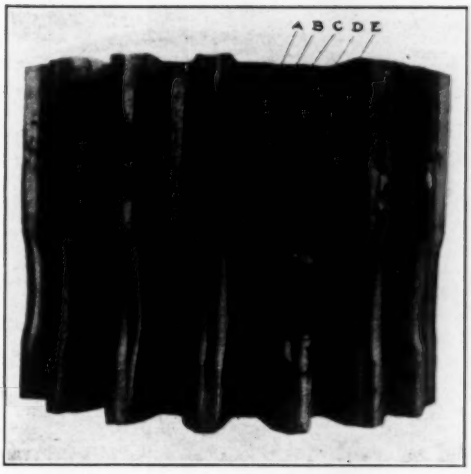

FIG. 3 16-TOOTH PINION BROKEN IN SERVICE

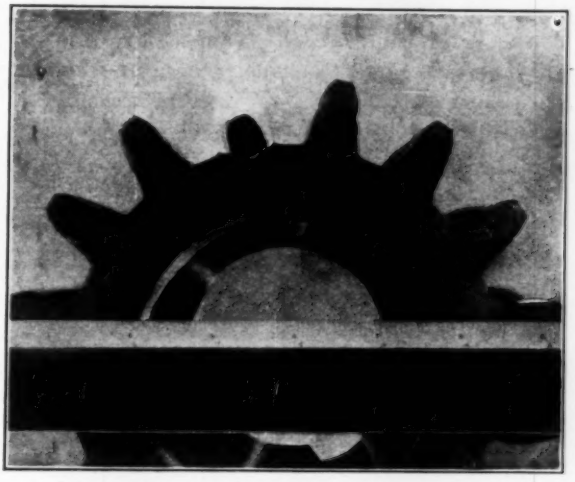

FIG. 4 END VIEW OF PINION SHOWN IN FIG. 3, SHOWING CRACKED TEETH

ture to cock that way, but in some cases it starts at the outside, probably caused by running a new pinion with a gear which had previously worn taper.

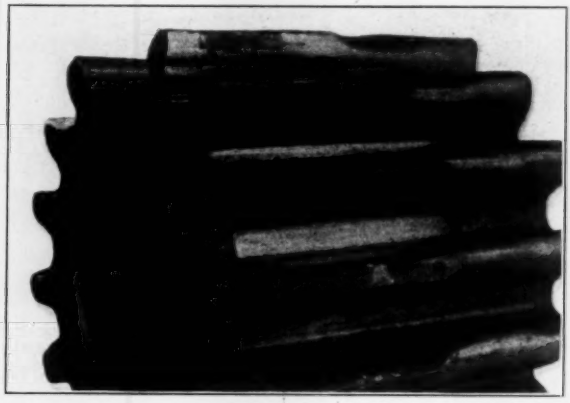

FIG. 5 THREE-QUARTER VIEW OF PINION SHOWN IN FIG. 3, SHOWING CRACKED TEETH

18 Furthermore the breakages cannot be attributed to so called crystallization or "gradual fracture," for in many instances the teeth are practically new, having broken after a inconsiderable mileage.

19 The necessity of having a high ratio between the theoretical working stress and the elastic limit thus becomes evident and justifies us in the adoption of the special means we have described to overcome the abnormal failures.

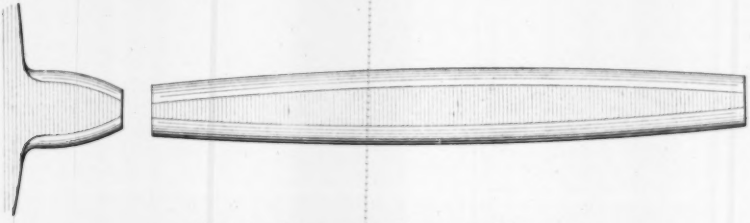

FIG. 6 SUGGESTED FORM OF TOOTH

20 The lack of alignment of gear and pinion above mentioned is probably one of the chief causes which necessitated the use of these special means, and we have therefore endeavored to find some other method of overcoming the difficulty. This led to the suggestion that

as we are actually obtaining only a short bearing on the tooth it would be better to assure this bearing being at the centre of the face rather than at the end, which might be accomplished by the use of a tooth something as shown in Fig. 6, in which the central portion is left the standard outline and the balance tapered off both ways toward the ends. No investigation has been made as to the practicability of cutting such a tooth or of its desirability, and it is suggested for consideration merely as one possible way of compensating for the lack of alignment occurring in railway motor gearing.

THE DEVELOPMENT OF A HIGH SPEED MILLING CUTTER, WITH INSERTED BLADES, FOR HIGH POWERED MILLING MACHINES

By Wilfred Lewis and Wm. H. Taylor

Member and Non-Member respectively

It has long been recognized that the process of milling is superior to any other for machining metal because the metal can be removed at a much higher rate of speed than by any other method, and as the operation of cutting is continuous, milling cutters being made up of a multiplicity of cutting edges, work can be machined at a much lower time cost. After the advent of high speed steel, builders of milling machines, especially those of the planer type, began to offer high-powered milling machines, and today it is a very common thing to see a milling machine whose spindle is driven by an individual motor of from 50 to 75 h.p. capacity, whose platen is driven by an independent motor of from 7 to 15 h.p. capacity and whose cross-rail is raised and lowered by an individual motor of from 3 to 5 h.p. capacity. While the milling machine has been developed to high power and high speed, the milling cutter has not advanced as rapidly; the user is thus confronted with a very unsatisfactory condition, the output of his milling machine being limited to cutters of inadequate capacity. This condition results from faults of design lying in the shape of the blade and method of fastening. This criticism applies to the inserted blade type of cutter, which by reason of its cheapness in first cost and maintenance has been universally adopted for heavy slab milling.

2 In 1892 George Brechtol developed a milling cutter of the inserted-blade type, in which the blades were made of carbon steel, properly shaped and secured. The results obtained at the time were considered extraordinary, being far in advance of those obtained

To be presented at the New York Meeting (December 1908) of The American Society of Mechanical Engineers.

by other types of carbon steel cutters. His cutter consisted of a malleable iron core, bored and keyseated, into whose body were planed 8 helical dove-tail grooves, considerably wider than the blades. The blades were bent around a cylinder to the desired helix. They were set inside a cylinder and properly spaced by means of blocks; the core was then set in position and soft metal poured, filling the dove-tail grooves and the spaces between the blades.

3 Mr. C. D. Peck was the first to develop successfully a milling cutter of the inserted-blade type for heavy slab milling, having high speed steel blades, helical in shape. The paramount feature in Mr. Peck's cutter was the helical shape of the blades. He planed helical slots, rectangular in shape, in a steel housing, inserted therein high-speed steel blades bent to fit the helix of the slot, and held them in a rigid position by means of wedges inserted at intervals between the front face of the blades and the side of the slot, filling the spaces between the wedges with soft metal. Mr. Peck designed and built a hand press for bending the blades, which consisted of a pair of lateral jaws encircling a shaft; one of them stationary, the other actuated by a lever and toggles so as to close upon the blade to form a helix.

4 His first cutter was built at the Pittsburg Works of the American Locomotive Company in 1905, and from tests made at the time, the results were far in excess of those obtained from other types of milling cutters with inserted blades, both in material removed in a specified time and in power consumed per cubic inch of material removed. He proceeded to build more cutters and put them into use, having conclusively demonstrated by continuous operation that the capacity of these cutters for removing metal was not only far greater than that of other types of inserted blade cutters, but in excess of the capacity of the modern type of high-powered milling machine.

5 Early in 1907 Mr. Peck had a conference with the manufacturers of the Taylor-Newbold Cold Saw, who were developing a milling cutter of the inserted-blade type, and the conference resulted in Mr. Peck turning his developments over to the Taylor-Newbold people. It was from this combination that the milling cutter about to be described was evolved. See Fig. 1.

6 Our primary investigations showed that there was no existing standard, or suitable rule, governing the construction of milling cutters with inserted blades, nor was there any record of exhaustive tests made to determine the most effective pitch, proper clearance

angles, or front slope and lip angles to be employed, and judging from past and present constructions in milling cutters of the inserted-blade type, the functional elements are merely arbitrary selections to suit individual tastes.

7 We also found the prevailing practice in constructing milling cutters with inserted blades to consist of cutting rectangular slots in a cylindrical housing, the slots lying in a plane angular to the axis. The angle from the axis at which the plane is set varies from 5 to 25 deg. according to the length and diameter of the housing. The blades are straight pieces of high speed steel, which are ground off after being inserted in the slots until a definite projection from the housing along the cutting edge is attained, after which clearance is obtained by "backing off."

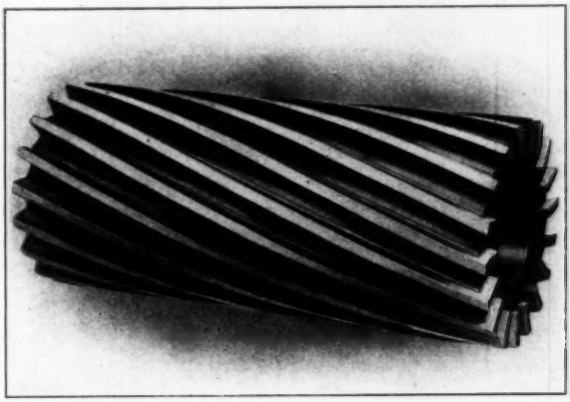

FIG. 1 HIGH SPEED INSERTED STEEL HELICAL BLADE MILLING CUTTER, 8 IN. DIAMETER, 3½ IN. BORE, 18 BLADES

8 The blades are held in position by various methods, such as clamps placed between each alternate pair of blades and drawn inward by screws; or by cutting grooves in the abutment between each alternate pair of blades and forcing the parted abutments by means of taper screws to clamp the blades on either side; or by driving wedges between the front face of the blade and the side of the slot; and by numerous other means.

9 We also found that much can be learned and applied to milling cutter blades from the development of the lathe and tool, especially the round nose tool; a fact totally ignored by the manufacturers

of milling cutters, who still try to accomplish the impossible,—that is, setting a straight blade in a slot with its front face in a plane angular to an axis, to develop thereby and maintain throughout its length the proper front slope and lip angle, and define a helix on the line of the cutting edge.

10 The first point which we considered in constructing the cutter to be described was the shape of the blade, with the following conclusions: To maintain a prescribed slope and lip angle throughout its entire length, the blade must be bent to form a helix, and by so shaping it all angles and the contour of both blades and slot would be constant over their entire lengths. Again, with the blades helical in shape, a continuous cutting edge with a constant lip angle would be maintained throughout any length of cutter.

11 The second point for consideration was the pitch or lead of the blade, and from experiments with various leads we found an effective angle for the helix to be about 20 deg. To facilitate computation, we adopted the formula, diameter $\times$ 9 = pitch, which would develop 19 deg., 15 min. as the angle of the helix.

12 The third point was the form of the grooves in the cutter blank. These had previously been planed approximately rectangular in section with a slight amount of undercutting to hold the blade and the wedges used for fastening it in place. It occurred to one of us however that this grooving of the cutter blank could be done better and faster by milling than by planing, and that an undercut groove might be produced at once by a saw set in a certain relation to the cutter blank. This suggestion was soon proved practicable, and although the groove so formed was not so easily fitted with a cutter blade on account of its curved sides, the curved sides gave the cutter a lip angle which was of great value in actual service. To form the blades accurately to the shape of the groove, it was necessary to design a bending machine of great power, capable of squeezing the blades at once to proper form not only as helices of correct pitch, but of correct curvature in a direction normal to the helix. See Fig. 2.

13 This machine was made to act in one way like the original hand-bending machine made by Mr. Peck, but in addition to the lateral jaws which closed on the blade to form the helix, it was provided with a cap which snapped quickly upon the lateral jaws, completely enclosing the blade being bent to proper form. Time is such an important element in the handling of high-speep steel that the value of this bending machine, which was quick enough to act before the temperature of the steel had fallen below the working point, can hardly be over-estimated.

14 Special furnaces for treating our blades were designed and built under the Taylor-White patents, but we are not at present so much concerned with the apparatus for manufacture as with the results obtained.

15 The fourth point for consideration was the method of securing the blades. Mr. Peck's experiments showed metal wedges to be neither satisfactory or economical as a means of securing the blades. While installed with ease they were exceptionally hard to remove when necessary to replace blades, owing to their tendency to imbed themselves in the housing. Furthermore, by driving them in at intervals along the high speed steel blade intermittent strains were

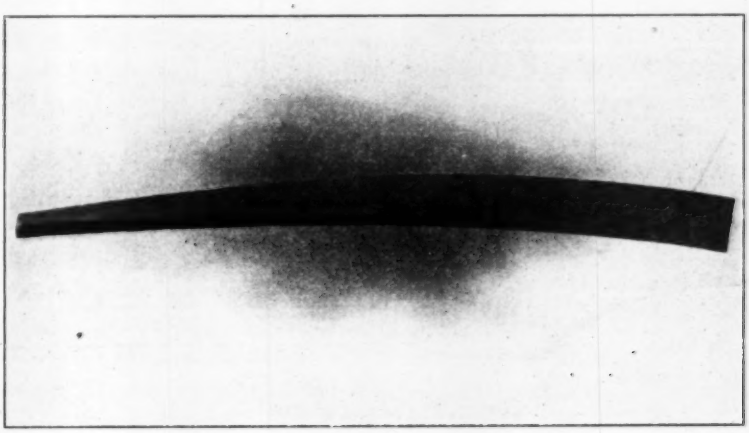

FIG. 2 HIGH SPEED STEEL BLADE BENT TO HELICAL SHAPE AND TREATED BY TAYLOR-WHITE PROCESS BEFORE INSERTION IN HOUSING

developed, causing the blade to crack and spall off at the point of contact between the wedge and blade when under cutting pressure. Other mechanical fastenings were debarred either by excessive cost or by inability to withstand vibration and remain rigid.

16 Experiments were made with various alloys until a proper combination was obtained, capable of flowing freely, cooling without shrinkage, withstanding great strains without crumbling and being removed quickly and economically.

17 A device was designed for compressing the alloy in the slots after it had been poured, and a device for removing the alloy when replacement of blades is necessary. With the alloy compressed in the slots we are able to secure the blades in an anchorage sufficiently

rigid so that the blades may be broken off by sheer force without affecting it. This form of construction enables us to produce a cutter of moderate diameter free from torsional strains and with a greater number of blades for a given diameter, and possessing a capacity in excess of the requirements of high-powered milling machines.

18 The milling cutter in question will be referred to here as a "Helical" rather than a "Spiral" cutter. The prevailing practice is to use these terms synonymously, but geometrically they are not even analogous; a helix is a line generated by progressive rotation

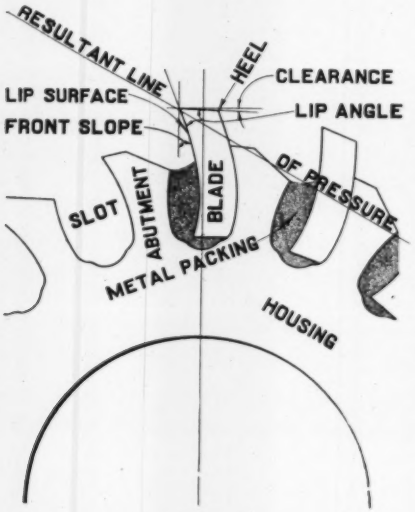

FIG. 3

of a point around an axis and equidistant from its center, while a spiral is generated by progressive rotation of a point around a fixed axis with a constantly increasing distance from the axis.

19 Fig. 3 shows the meaning of the following terms: housing-body of cutter; abutment—support of blade; metal packing—anchorage; slot; blade; heel of blade; clearance angle; lip angle; front slope; lip surface and resultant line of pressure. The side slope is defined by the angle of the helix.

20 Fig. 4 shows how the chip is partly torn and partly sheared from body of forging, the cutting edge of blade not being under heavy pressure. Our experiments have shown conclusively that the closer the center of pressure of the chip is to the cutting edge, the greater

its intensity, and the generated heat is largely concentrated towards the cutting edge, where the sectional area to carry it off is much less. Then again the frictional heat generated by lack of back slope becomes so great as to cause cohesion between the chip and the cutting edge. Under heavy pressure we have seen a compact of chip and blade which was virtually inextricable, so that when disunited from the blade the cutting edge still adhered to the chip. To eliminate

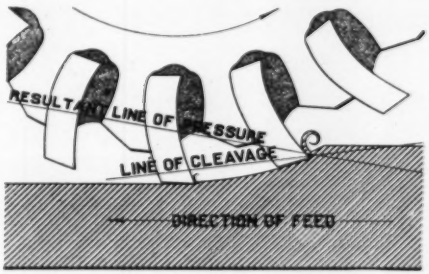

FIG. 4

this condition where straight blades are employed, the practice is to set the front face of the blade slightly back of a radial plane (see line R, Fig. 5) to assure a front slope. It is well known that the absence of front slope eliminates any proclivity which the lip may have to develop a line of cleavage and thereby throw the pressure of the chip back on the lip surface from the cutting edge.

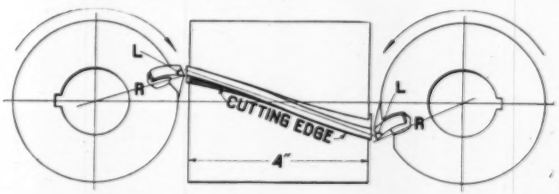

FIG. 5

21 Fig. 5 shows a constant lip angle L throughout the entire length of the blade, which is set at an angle of 20 deg. to an axial plane. this angle remaining constant throughout any length by reason of the blade's curvature.

22 Fig. 6 shows a varying lip angle L from maximum at line R1 to minimum at line R2 in a straight blade set at an angle of 20 deg. to an axial plane. This condition limits the length of blades.

23 In Fig. 7 a straight blade is set in a plane radiating from the axis, designated by line R1, and by carrying it across the face of the housing at an angle set at 20 deg. to an axial plane. You may observe the development from no front slope to a positive front slope, designated by the letter S. In milling, a blade with this irregularity in front slope causes the cutter to drag on one side and gouge on the other. Blades of this type cause excessive vibration to the cutter, due to the varying angle of the front slope, and necessarily consume more power. It has been contended that by nicking the blades to alleviate

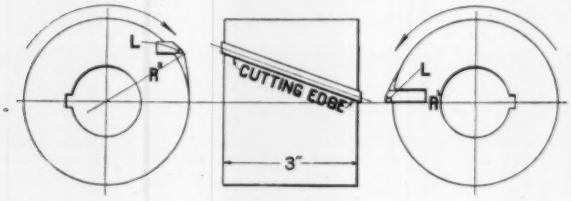

FIG. 6

the tendency of the blade to gouge and drag, and to define a more even pressure throughout the line of the cutting edge, less power would be required to drive the cutter; the assumption being that less power would be consumed in breaking up the chip than would be required by a continuous cutting edge producing a continuous chip.

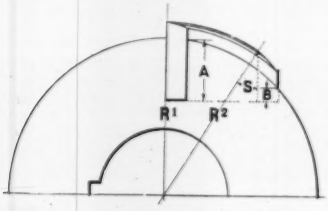

FIG. 7

24 We differ with this contention, having demonstrated conclusively by experiments that the initial fault is caused by absence of front slope, and that an undesirable feature is developed by nicking the blades. The blade behind the nick whose cutting edge covers the gap formed by a nick in the blade preceding must accept double the feed; this causes chatter and produces an uneven machined surface.

25 It is also a fact that when a rectangular slot is cut in the face of a housing at an angle of 20 deg. to an axial plane, its depth becomes gradually less as it progresses across the face, until it reaches a vanishing point designated by lines A and B in Fig. 7. This condition not only limits the width of the cutter and necessitates a housing of large diameter when a medium width of cutter is required, but develops an anchorage possessing the required strength and rigidity at one side and decreasing in proportion to the length of blades at the other. Where a wide face is required it is necessary to construct a cutter composed of a number of sections.

26 Fig. 8 illustrates a cutter with straight inserted blades made up in sections, each alternate blade overlapping the blades in the opposite section, so as to obtain the desired width of face. The sec-

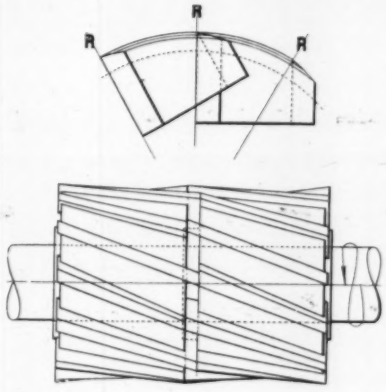

FIG. 8

tions are so set that the cutting edge forms a continuous line. Above the cutter is a diagram showing the relation of the blades in one section to those in the other.

27 Too much stress cannot be laid on the use of a lubricant during the process of milling. A copious stream of lubricant falling at slow velocity should be thrown directly upon the chip at the point of removal. Heat generated by the pressure of the chip is the chief cause for wear, and if allowed to become too great it will soften the lip surface of the blades and cause them to crumble or spall off. An ample supply of lubricant during the milling operation carries off the heat, materially lessening the dulling of the cutting edges.

28 From our experiments and those of others it has been conclusively shown that a gain of 33 per cent in the cutting speed in milling

steel and wrought iron is made by throwing a heavy stream of lubricant upon the cutter and along its entire face; and a gain of 15 per cent in milling cast iron. The piping for conveying the lubricant to the milling cutter should be arranged with nozzles spaced about 4 in.

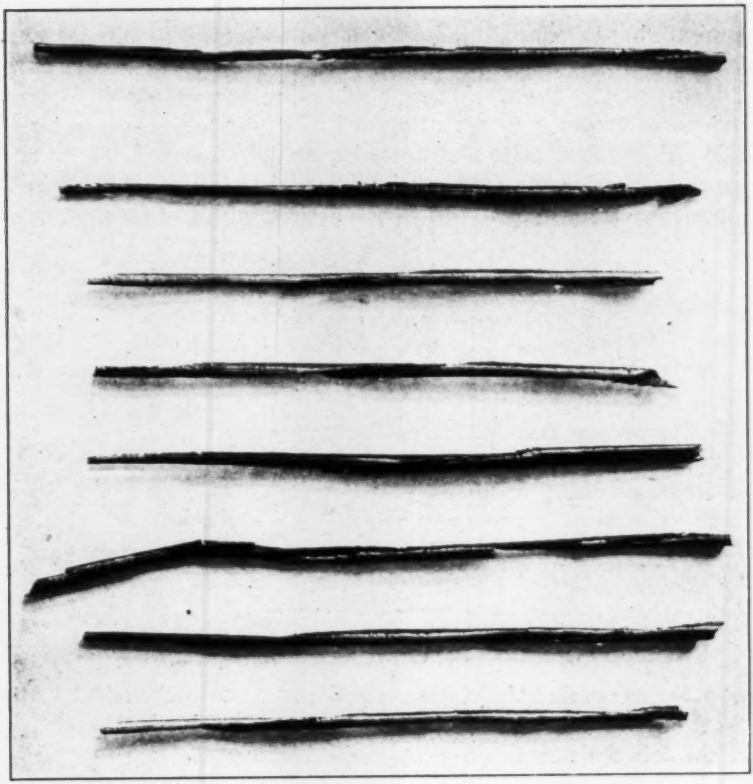

FIG. 9 STEEL CHIPS MILLED FROM 0.57 PER CENT CARBON STEEL FORGING 18 IN. WIDE AT THE RATE OF 47½ CU. IN. PER MIN.
THESE CHIPS VARY FROM 14 TO 16½ IN. IN LENGTH

apart, in sufficient number to cover the face of the cutter. The main supply pipe to the nozzles should be large enough to supply each nozzle with from two to three gallons of lubricant per minute. The general arrangement, size of supply pipe and number of nozzles will

be regulated by the width of the machine and the nature of the work to be done.

29 In the more recent designs of slab milling machines due consideration has been given to lubrication. The platen is drained by

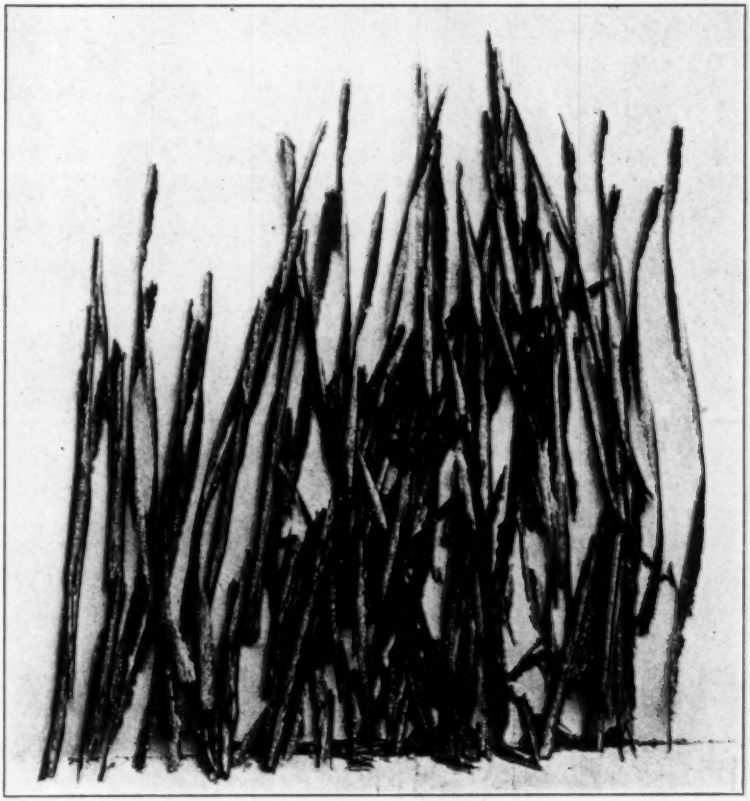

FIG. 10 CAST IRON CHIPS MILLED FROM A SURFACE 14 IN. WIDE.

THESE CHIPS WERE REMOVED AT THE RATE OF 105 CU. IN. PER MIN. AND AS A WHOLE ARE STRAIGHT AND CYLINDRICAL IN SHAPE. THEY AVERAGE ABOUT 12 IN. IN LENGTH AND ARE STRONG ENOUGH TO SUPPORT THEMSELVES WHEN STANDING ON END.

gravity to a tank located at the side of the machine, from which the lubricant is raised by a pump to a reservoir formed by the cross piece at the top of the housings, to which is attached the nozzle bracket that conveys the lubricant to the required point of gravity.

FIG. 11 BEMENT-MILES HIGH-POWERED MILLING MACHINE

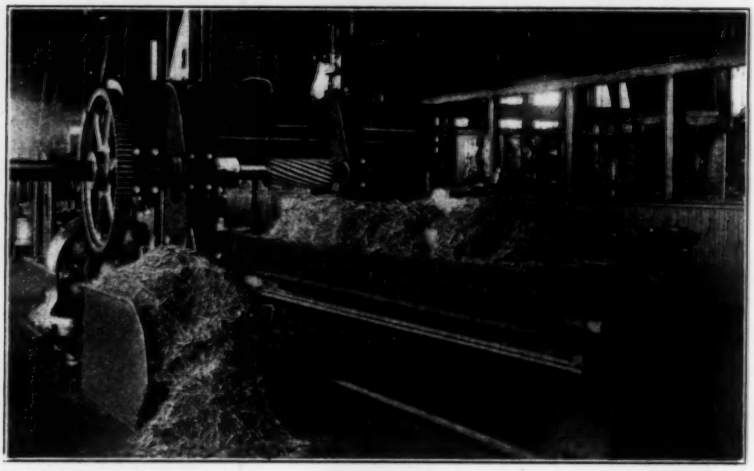

FIG. 12 MACHINE AT COMPLETION OF CUT

30 In the tabulated tests the amount of power consumed by the milling machine will be given as read from the voltmeter and ammeter. Dynamometer readings of the actual power exerted at the spindle, or the power consumed by friction in the gearings and bearings, will not be given consideration; the user being most interested in the commercial readings that indicate the power for which he has to pay.

31 Credit is due the Bement-Miles Works of the Niles-Bement-Pond Company, of Philadelphia, for the use of various types of high-powered milling machines, and valuable assistance in carrying out experiments and tests.

32 The authors regret that in order to present the above data at the coming meeting it has been impossible to complete their experiments.

TABLE 1 SLAB MILLING CAST IRON, TAYLOR-NEWBOLD HIGH-SPEED STEEL MILLING CUTTER, 8-IN. DIAMETER, 18-IN. FACE 18 INSERTED BLADES. TEST MADE AT BEMENT-MILES WORKS, JULY 13, 1908

MACHINE USED: 42-IN. BEMENT-MILES MILLING MACHINE
DRIVING MOTOR: WESTINGHOUSE DIRECT-CURRENT CONSTANT SPEED TYPE 40-H.P. AT 220 VOLTS, 153 AMPERES
MATERIAL CUT: CAST IRON TEST BLOCK, 15 IN. WIDE, 36 IN. LONG

Number of test	CUT							Duration of test	SPEED OF CUTTER		ELECTRICAL READINGS			H.p. per cubic inch removed
	FEED		Depth Inches	Width Inches	MATERIAL REMOVED				R.p.m.	Feet per minute	DRIVING MOTOR			
	Table advance per minute Inches	Advance per blade Inches			Cubic Inches per minute	Pounds per minute	Pounds per hour				Amperes	Volts	H.p.	
1	3½	.00636	½	15	26.25	6.83	410.13	10m. 2s.	25	53½	150	200	40.21	.53
2	5¼	.01332	½	15	44.06	11.47	688.38	6m. 8s.	24½	51⅓	150	220	44.23	1.00
3	7½	.01736	½	15	56.25	14.74	737.33	4m. 48s.	24	50¼	160	208	44.70	0.794
4	6¼	.01361	½	15	45.93	11.96	717.72	5m. 5s.	25	53½	175	208	48.79	1.060
5	7⅔	.01722	½	15	58.12	15.13	908.14	4m. 39s.	25	53½	225	200	60.32	1.030
6	7⅜	.01781	11/16	15	50.70	13.20	792.13	4m. 53s.	23	48	260	204	70.96	1.390
7	7⅜	.01781	11/16	15	50.70	13.20	792.13	3m. 39s.	23	48				
8	8	.01851	11/16	15	82.50	21.48	1288.98	1m. 15s.	24	50¼	240	213	68.52	.830
9	7	.01760	1	15	105.00	27.34	1640.52	3m. 17s.	22	47	350	190	89.14	.849

TABLE 2 SLAB MILLING STEEL, TAYLOR-NEWBOLD HIGH-SPEED STEEL MILLING CUTTER, 8-IN. DIAMETER, 18-IN. FACE, 18 INSERTED BLADES. TEST MADE AT BEMENT-MILES WORKS, JULY 14, 1908

MACHINE USED: 42-IN. BEMENT-MILES MILLING MACHINE
DRIVING MOTOR: WESTINGHOUSE DIRECT-CURRENT CONSTANT SPEED TYPE 40 H.P. AT 220 VOLTS, 153 AMPERES
MATERIAL CUT: 30 PER CENT CARBON STEEL TEST BLOCK 18 IN. WIDE, 20 IN. LONG

Number of test	Cut							Duration of test	Speed of Cutter		Electrical Readings			H.p. per cubic inch removed
	Feed		Depth Inches	Width Inches	Material removed						Driving motor			
	Table advance per minute Inches	Advance per blade Inches			Cubic Inches per minute	Pounds per minute	Pounds per hour		R.p.m.	Feet per minute	Amperes	Volts	H.p.	
1	11½	.01785	3/16	18	31.64	8.96	537.81	1m. 47s.	35	73½	190	220	56.03	1.77
2	11½	.01785	3/16	18	63.28	17.92	1075.62	0m. 11s.	35	73½				
3	6⅜	.01041	¼	18	43.03	12.19	731.44	1m. 30s.	34	71	300	195	78.41	1.82
4	7	.01080	¼	18	47.25	13.38	801.15	1m. 26s.	36	75½	400+	180	96.51	2.04
5	7	.00925	¼	18	47.25	13.38	801.15	2m. 51s.	42	88	370	187	92.74	1.96

TABLE 3 MILLING CHANNELS, TAYLOR-NEWBOLD HIGH-SPEED STEEL MILLING CUTTER 4¾-IN. FACE, 8-IN. LISTED DIAMETER, 8¼ IN. ACTUAL DIAMETER, 18 IN. INSERTED BLADES, 3½-IN. BORE. TEST MADE AT BEMENT-MILES WORKS, OCTOBER 20, 1908

MACHINE USED: 42-IN. BEMENT-MILES MILLING MACHINE
DRIVING MOTOR: WESTINGHOUSE DIRECT-CURRENT CONSTANT SPEED TYPE 40-H.P. AT 220 VOLTS, 153 AMPERES
MATERIAL CUT: 35 PER CENT CARBON STEEL FORGING

Number of test	Cut							Duration of test	Speed of Cutter		Electrical Readings			H.p. per cubic inch removed
	Feed			Material removed							Driving motor			
	Table advance per minute Inches	Advance per blade Inches	Depth Inches	Width Inches	Cubic Inches per minute	Pounds per minute	Pounds per hour		R.p.m.	Feet per minute	Amperes	Volts	H.p.	
1									39	82.95	35	220	10.32	
2	6⅞	.01320	1/16	4¾	9.40	2.66	159.78	3m. 25s.	37	78.69	85	200	22.74	2.42
3	1 1/16	.00307	1	4¾	8.48	2.40	144.14	7m. 12s.	35	74.44	95	185	23.56	2.77
4	1½	.00231	1	4¾	6.56	1.86	111.51	4m.	36	76.57	85	193	21.85	3.33
5	2⅛	.00394	1 1/16	4¾	15.07	4.27	256.15	2m. 40s.	37	78.69	113	200	30.29	2.00
6	3⅛	.00544	1 1/16	4¾	20.81	5.89	353.72	1m. 56s.	37	78.69	135	195	35.28	1.69
7	5	.00750	1 1/16	4¾	28.71	8.13	488.01	1m. 24s.	37	78.69	200	190	50.93	1.77
8														

EMPLOYMENT BULLETIN

The Society has always considered it a special obligation and pleasant duty to be the medium of securing better positions for its members. The Secretary gives this his personal attention and is most anxious to receive requests both for positions and for men available. Notices are not repeated except upon special request. Copy for notices in this Bulletin should be received before the 15th of the month. The list of men available is made up of members of the Society and these are on file, with the names of other good men not members of the Society, who are capable of filling responsible positions. Information will be sent upon application.

POSITIONS AVAILABLE

038 Water Works superintendent in small town, New York State.

MEN AVAILABLE.

166 Power plant design or steel car design in or near New York City.

167 Member, thirty-five, graduate mechanical engineer, five years experience as instructor and assistant professor of mechanical engineering in a well known university; at present and for three and a half years past, continuous successful experience as a mechanical engineer, acting as assistant superintendent, superintendent of construction, chief draftsman, designer of tools, fixtures, buildings, special machines, gasoline, kerosene, alcohol engines, cement specialties machines; developed cost-keeping and cost-reducing systems. Desires position as professor or assistant professor where research work may be carried on.

168 Practical machinist, graduate mechanical engineer, iron and ceramic chemist, one year experience after graduation as salesman of engineering specialties, two years experience as engineer and chemist in large foundry for enameled iron.

169 Mechanical engineer, college graduate, desires to make change; twelve years experience drawing room, shop, salesman, and road experience; would like position as salesman on road for some manufacturing concern selling engines, boilers, machinery, pumps or power plant specialties.

170 Member, thirty-five, married, fourteen years broad experience in design and construction of power plants, water works, machine shops, cement mills and industrial plants, wants position with consulting engineer or consulting and contracting engineering company, or large manufacturing company as resident engineer or superintendent of construction; good executive, designer and handler of men.

171 Mechanical engineer with shop experience as molder, pattern-maker, machinist and draftsman; has been engaged on new design and revision of existing designs for more economical production, also devising shop methods and putting them in operation, and at present serving as superintendent of manufacturing plant turning out a line of medium weight machinery; desires position in which such experience would be of value.

172 Mechanical engineer, Civil Engineer, University of Virginia, twenty-three years experience in various branches of engineering, nine years designing and installing hydro-electric power plants. Position designing or installing hydro-electric plants.

173 Mechanical engineer, technical graduate, one year additional in Electrical Engineering, experienced in shop, testing, and office work; now holding responsible position in public service, desires to enter manufacturing or contracting concern.

174 Member, graduate of Mass. Institute of Technology, ten years practical experience in shop work involving machine, foundry, and boiler practice; desires opportunity to take charge of shop management. Can give best references as to character and capability.

175 Mechanical engineer with very extensive experience in connection with large power station design and construction, also general engineering, will soon be available.

CHANGES IN MEMBERSHIP

CHANGES OF ADDRESS

ADAMS, Julius Leroy (1900), 155 Bute St., Norfolk, Va.

BERRY, Edgar H. (1905; 1907), 248 N. 20th St., East Orange, N. J.

BOLLES, Frank G. (Associate, 1901), Tungstolier Co., 227 Fulton St., New York, N. Y., and *for mail*, 188 Market St., Newark, N. J.

BRENNER, Wm. H. (1897), British Columbia Mfg. Co., New Westminster, B. C., Canada.

BURTON, Isaac Francis (1908), Asst. Supt., Victor Talking Mch. Co., 5219 Walnut St., Philadelphia, Pa.

COMSTOCK, Chas. Warren (Associate, 1906), Pres., The Comstock Brass Fdy. Co., 2539 E. 79th St., Cleveland, O.

DENSMORE, Edward D. (Associate, 1898), Cons. Engr., Densmore & LeClear, 15 Exchange St., and 52 River St., Boston, Mass.

DICKIE, Geo. W. (1892), Manager, 1895–1898, Room 722, Marvin Bldg., 24 California St., and 796 Asbury St., San Francisco, Cal.

DURAND, Nelson Crawford (1889), Mgr. of Sales, Edison Business Phonograph Co., Orange, and *for mail*, Washburn Pl., Caldwell, N. J.

EBERHARDT, Henry J. (1902; Associate 1906), Secy., Newark Gear Cutting Mch. Co., 66 Union St., Newark, N. J.

EILERS, Carl Emrich (1890; 1904), Am. Smelting and Refining Co., 165 Broadway, and *for mail*, 320 Central Park W., New York, N. Y.

GAY, Harry (Associate, 1907), 1219 Grand Central Sta., and *for mail*, 200 Claremont Ave., New York, N. Y.

GEBHARDT, George F. (1904), Prof. Mech. Engrg., Armour Inst. of Tech., and 5329 Kimbark Ave., Chicago, Ill.

GOWIE, William (1905), Supt., Scott Iron and Steel Co., Carnegie, Pittsburg, Pa.

HASKELL, George O. (Junior, 1899), V. P., Fidelity Cotton Oil & Fertilizer Co., Houston, Texas.

HERR, Herbert T. (1907), 1529 Gaylord St., Denver, Colo.

HIGGINS, George Frederick (1886; 1906), Cons. Engr., 63 Botolph St., Melrose Highlands, Mass.

HILES, Elmer K. (1900; 1906), 803 Fulton Bldg., and *for mail*, 327 Stratford St., Pittsburg, Pa.

HOGLE, Milton W. (Junior, 1901), 128 Linden St., Rochester, N. Y.

HUTCHINS, Harry C. (Junior, 1905), Structural Draftsman, Public Service Com., 154 Nassau St., New York, N. Y.

KOON, Sidney Graves (Junior, 1905), 166 Rogers Ave., Brooklyn, N. Y.

KREUTZBERG, Otto August (Associate, 1904), Secy. William B. Hough Co., 1003 First Ntl. Bank Bldg., and 38 Roslyn Pl., Chicago, Ill.

LARSON, Charles J. (1907), Ch. Engr., Union Elec. Co., Dubuque, Ia.

LINDQUIST, Edward W. (1902; 1905), Allis-Chalmers Co., 599 Mission St., San Francisco, Cal.

LORD, John E. (1904), 6 N. 10th St., Hamilton, O.

LUHR, Otto (1907), Instr. Mech. Engrg., Wahi-Hennis Inst., and *for mail* 434 Potomac Ave., Chicago, Ill.

McKECHNIE, Robert R. (1897), Mech. Engr., 2170 E. 79th St., Cleveland, O.

MERSEREAU, Theodore T. (1905), Eastern Sales Agent, The Casey-Hedges Co., 100 William St., New York, and *for mail*, 448 56th St., Brooklyn, N. Y.

MIDGLEY, Frederick W. (Junior, 1904) Ch. Draftsman Otis Elevator Co., and *for mail*, 23 Florence St., Yonkers, N. Y.

OSTRANDER, Allen Edward (Associate, 1905), Asst. Mech. Engr., Am. Car and Fdy. Co., 165 Broadway, New York, N. Y., and 33 18th Ave., Paterson, N. J.

PEEK, George Meredith (1892; 1900), Pittsboro, N. C.

PENNINGTON, James H. (1902), Mech. Engr., Bettendorf Axle Co., 42 Broadway, New York, N. Y., and *for mail*, 9 Water St., Perth Amboy, N. J.

PRATT, James Alfred (Associate, 1908), Williamson School P. O., Delaware Co., Pa.

RAMSAY, James D. (1890), 515 Church St., Ann Arbor, Mich.

ROSE, William Holliday (Junior, 1901), Park Ave. Hotel, New York, N. Y.

SCHERR, Frederick, Jr. (Associate, 1907), The Fajardo Sugar Co., Fajardo, Porto Rico.

SETHMAN, George H. (Junior, 1899), Mech. and Elec. Engr., 1540 Grant Ave., Denver, Colo.

SHAW, Arthur D. (Associate, 1905), Engr., Link Belt Co., Nicetown, Philadelphia, and *for mail*, 318 Hansberry St., Germantown, Philadelphia, Pa.

SMITH, T. Carpenter (1886), Mech. Engr., 220 S. Broad St., Philadelphia, Pa.

SPIES, Albert (1884), Editor, The Electrical Record, and Steam, 114 Liberty St., New York, N. Y., and *for mail*, 40 Glenwood Ave., Jersey City Heights, N. J.

STURGESS, John (1901), Genl. Mgr., The Lombard Replogle Engrg. Co., and *for mail*, 9 Arch St., Akron, O.

SWAN, John Joseph (Junior, 1899), Cynwyd, Pa.

THOMPSON, Edward P. (1884), M. E., 34 E. 32d St., New York, N. Y.

THULLEN, L. H. (1905), Mech. and Elec. Engr., 622 W. 137th St., New York, N. Y.

VAN BUSKIRK, Henry C. (1906), Supt. M. P., Colo. and Southern Ry., and *for mail*, 1121 Clarkson St., Denver, Colo.

WHITING, S. B. (1880), Manager, 1880-1882; V. P., 1882–1883; 11 Ware St., Cambridge, Mass.

WINTER, Oscar (1906), Hotel Euclid, Cleveland, O.

WINTERROWD, William H. (Junior, 1907), Round-house Foreman, L. E. A. and W. Ry., and *for mail*, P. O. Box 201, Alliance, O.

WRIGHT, Wm. Quimby (1908), Member, Wright-Plummer Co., Marvin Bldg., 24 California St., San Francisco, Cal.

YORK, Robert (Junior, 1901), V. P., York-Browning Lumber Co., Memphis, Tenn.

NEW MEMBERS

SEAMAN, Edwin H. (Junior, 1908), Fire Ins. Insp., American Lloyds, 45 Cedar St., New York, and *for mail*, Box 66, Wantagh, Nassau Co., N. Y.

WYNKOOP, Hubert S. (1908), Cons. Mech. and Elec. Engr., 1574 50th St., Brooklyn, N. Y.

RESIGNATIONS

AUCHINCLOSS, Wm. S.
BLANCHARD, Gilbert W.
BROWN, Carlton E.
CONKLIN, Millard Thorn
DUDLEY, Samuel W.
GORDON, Alex.
GREEN, Morris M.
GUTHRIE, Edward Buckingham
HANNA, Elmer E.
KING, Charles C.
KRETSCHMER, F. G.
LINZEE, A. C.
LOVELAND, John Winthrop
MACKINTOSH, Frederick
MASSIE, James H.
PORTER, Arthur T.
PRITCHARD, W. S.
WARE, Justin A.
WHITNEY, Marshall Long
WICKHORST, Max Henry

DEATHS

CANET, Gustav
PERRINE, Frederic A C

COMING MEETINGS

AMERICAN ASSOCIATION FOR THE ADVANCEMENT OF SCIENCE
Week of December 28, annual meeting at Johns Hopkins University, Baltimore, Md. Secy., L. O. Howard, Washington, D. C.

THE AMERICAN INSTITUTE OF ARCHITECTS
December 15–17, 1908, annual convention, Washington, D. C. Secy., Glenn Brown, The Octagon.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS
December 11, 33 W. 39th St., New York, 8 p.m. Secy., R. W. Pope.

AMERICAN GEOGRAPHICAL SOCIETY
November 24, December 22, 29 W. 39th St., New York, 8 p.m. Secy., A. A. Raven.

AMERICAN SOCIETY OF CIVIL ENGINEERS
December 2–16, New York. Secy., Chas. W. Hunt, 220 W. 57th St.

AMERICAN SOCIETY OF MECHANICAL ENGINEERS
December 1–4, annual meeting, 29 W. 39th St., New York, 8 p.m. Secy., Calvin W. Rice.

AMERICAN SOCIETY OF REFRIGERATING ENGINEERS
November 30–December 1, annual meeting, 29 W. 39th St., New York. Secy., W. H. Ross, 154 Nassau St.

ASSOCIATION OF TRANSPORTATION AND CAR ACCOUNTING OFFICERS
December 9, Cincinnati, O. Secy., G. P. Conard, 24 Park Pl., New York.

BLUE ROOM ENGINEERING SOCIETY
December 3, 29 W. 39th St., New York, 8 p.m. Secy., W. D. Sprague, 100 N. Moore St.

BOSTON SOCIETY OF CIVIL ENGINEERS
December 16, monthly meeting, Tremont Temple. Secy., S. E. Tinkham, 60 City Hall.

CANADIAN SOCIETY OF CIVIL ENGINEERS
November 26, electrical section, December 3, mechanical section, December 10, business meeting, December 17, mining section, 413 Dorchester St., W., Montreal, Que. Secy., Prof. C. H. McLeod.

CAR FOREMEN'S ASSOCIATION OF CHICAGO
December 14. Secy., Aaron Kline, 326 N. 50th St.

CENTRAL ASSOCIATION OF RAILROAD OFFICERS
December 14, Kansas City. Secy., J. H. Ashley, Gumbel Bldg.

CENTRAL ASSOCIATION OF RAILROAD OFFICERS
December 10, Toledo, O. Secy., H. M. Elliott.

CENTRAL RAILWAY AND ENGINEERING CLUB OF CANADA
December 15, Rossin House, Toronto, Ont. Secy., C. L. Worth, Room 409, Union Station.

CANADIAN RAILWAY CLUB

December 1, Montreal, Que. Secy., Jas. Powell, Grand Trunk Ry.

CLEVELAND ENGINEERING SOCIETY

December 8, monthly meeting, Caxton Building. Secy., Joe C. Beardsley.

COLORADO SCIENTIFIC SOCIETY

December 5, Denver. December 19, annual meeting. Secy., Dr. W. A. Johnston, 801 Symes Bldg.

ENGINEERING ASSOCIATION OF THE SOUTH

December 15, monthly meeting, Nashville Section, Carnegie Library Bldg. Section Secy., H. H. Trabue, Berry Block, Nashville.

ENGINEERING SOCIETY OF THE STATE UNIVERSITY OF IOWA

December 1, monthly meeting, Iowa City, Ia. Dean, Wm. G. Raymond.

ENGINEERS AND ARCHITECTS' CLUB OF LOUISVILLE, KENTUCKY

December 2, monthly meeting, 303 Norton Bldg. Secy., Pierce Butler.

ENGINEERS' CLUB OF BALTIMORE

December 5. Secy., R. K. Compton, City Hall.

ENGINEERS' CLUB OF CENTRAL PENNSYLVANIA

December 1, monthly meeting, Gilbert Bldg., Harrisburg. January 5, 1909, annual meeting. Secy., E. R. Dasher.

ENGINEERS' CLUB OF CINCINNATI

December 17, 25 E. 8th St. Secy., E. A. Gast, P. O. Box 333.

ENGINEERS' SOCIETY OF MILWAUKEE

December 9, monthly meeting. June 9, 1909, annual meeting. Secy., W. Fay Martin, 396 Bradford Ave.

EXPLORERS CLUB

December 4, 29 W. 39th St., New York, 8.30 p.m. Secy., H. C. Walsh.

GENERAL SUPERINTENDENTS' ASSOCIATION

December 16, Chicago, Ill. Secy., H. D. Judson, C. B. & Q. R. R., Chicago.

ILLUMINATING ENGINEERING SOCIETY

December 10, New York section, 29 W. 39th St., 8 p.m. Secy., P. S. Millar.

IOWA RAILWAY CLUB

December 11, Des Moines. Secy., W. B. Harrison, Union Sta.

LOUISIANA ENGINEERING SOCIETY

December 14, monthly meeting, 323 Hibernia Bldg., New Orleans. January 9, 1909, annual meeting, New Orleans. Secy., L. C. Datz, 323 Hibernia Bldg.

MASSACHUSETTS STREET RAILWAY ASSOCIATION

December 9, Boston. Secy., Charles S. Clark, 70 Kilby St.

MUNICIPAL ENGINEERS OF THE CITY OF NEW YORK

November 25, December 23, 29 W. 39th St., 8:15 p.m. Secy., C. D. Pollock. Paper: Waterproofing and Engineering Problem, Myron H. Lewis.

NATIONAL SOCIETY FOR PROMOTION OF INDUSTRIAL EDUCATION

November 19–21, annual meeting at Atlanta, Ga. Secy., James P. Haney, 546 Fifth Ave., New York. Papers: Industrial Education as an Essential Factor in our National Prosperity, James Wilson, E. E. Brown, Carroll D. Wright; Industrial Training through the Apprentice System, E. P. Bullard; Promotion of Industrial Education by Means of Trade Schools, George N. Carman, J. M. Shrigley; Moral and Material Benefits of Industrial Educa-

tion to the Nation, G. G. Jordon, Mrs. B. B. Munford; Industrial Education in the Public Schools, Thomas M. Balliet, L. D. Harvey.

NEW ENGLAND RAILROAD CLUB

December 8, monthly meeting, Boston, Mass. Secy., Geo. H. Frazier, 10 Oliver St., Boston.

NEW ENGLAND STREET RAILWAY CLUB

December 24, American House, Boston, Mass. March 25, 1909, annual meeting. Secy., John J. Lane, 12 Pearl St.

NEW ENGLAND WATER WORKS ASSOCIATION

December 9, regular meeting. Secy., Willard Kent, Tremont Temple, Boston, Mass.

NEW YORK RAILROAD CLUB

November 20, December 18, 29 W. 39th St., Paper, November: A Search for those Elements the Proper Combination of which Constitutes the Successful Railway Official, W. J. Harahan. Secy., H. D. Vought, 95 Liberty St.

NEW YORK SOCIETY OF ACCOUNTANTS AND BOOKKEEPERS

November 24, December 1, 8, 15, 22, 29 W. 39th St., 8 p.m. Secy., T. L. Woolhouse.

NEW YORK TELEPHONE SOCIETY

December 15, 29 W. 39th St., 8 p.m. Secy., T. H. Laurence, 63 Irving Pl., New York.

NORTHERN RAILWAY CLUB

November 28, Commercial Club Rooms, Duluth, Minn. Secy., C. L. Kennedy.

NORTH WEST RAILWAY CLUB

December 15, St. Paul, Minn. Secy., T. W. Flannagan, Soo Line, Minneapolis, Minn.

OHIO SOCIETY OF MECHANICAL, ELECTRICAL AND STEAM ENGINEERS

November 20–21, Toledo. Secy., David Gaehr, 1135 Schofield Bldg., Cleveland. Papers: The New Engineering Laboratory, Profs. F. C. Caldwell and E. A. Hitchcock of Ohio State University; description of The Columbus Municipal Electric Light Plant, G. H. Gamper; Underground Heat Insulation of Steam Piping, H. K. Gillette; description of the Toledo Filtration Plant and Waterworks, Supt. Taylor; Refrigeration, Thos. Baker.

PACIFIC NORTHWEST SOCIETY OF ENGINEERS

December 5, monthly meeting, Lowman Bldg., Seattle, Wash. Secy., Arthur H. Dimock, City Engineer's Office. Paper: Sluicing Operations on Topographers Hill, E. W. Commings; December 5, Seattle Sea Wall, D. W. McMorris.

PROVIDENCE ASSOCIATION OF MECHANICAL ENGINEERS

November 24, December 22, monthly meetings, 48 Snow St. June 22, 1909, annual meeting. Secy., T. M. Phetteplace.

PURDUE MECHANICAL ENGINEERING SOCIETY

November 25, Purdue University, Lafayette, Ind., 6:30 p.m. Secy., L. B. Miller.

RAILWAY CLUB OF PITTSBURGH

November 27, annual meeting, December 25, monthly meeting, Monongahela House, Pittsburg, Pa., 8 p.m. Secy., J. D. Conway, Gen. Office, P. & L. E. R. R.

RENSSELAER SOCIETY OF ENGINEERS

November 20, December 4, 257 Broadway, Troy, N. Y. Secy., R. S. Furber.

RICHMOND RAILROAD CLUB

December 14. Secy., F. O. Robinson, C. & O. Railway.

ROCHESTER ENGINEERING SOCIETY

December 11, monthly meeting. Secy., John F. Skinner, 54 City Hall.

SCRANTON ENGINEERS' CLUB

December 17, Board of Trade Building. Secy., A. B. Dunning.

SHORT LINE RAILROAD ASSOCIATION

December 7, New York. Secy., Cromwell G. Macy, Jr., Nantucket Central Railroad, 257 Broadway.

SOCIETY OF NAVAL ARCHITECTS AND MARINE ENGINEERS

November 19–20, annual meeting, 29 W. 39th St., New York. Secy., Wm. J. Baxter, Bldg. No. 6, Navy Yard, Brooklyn. Papers: The War Eagle, C. H. Cramp; Practical Methods of Conducting Trials of Vessels, E. A. Stevens; Influence of Midship Section on Resistance of Ships, D. W. Taylor; Longitudinal Displacement and Effect on Resistance, H. C. Sadler; Further Analysis of Propeller Experiments, C. H. Crane; Deviation of Compass aboard Steel Ships, its Avoidance and Correction, L. H. Chandler; Free Water Ballast, T. G. Roberts; Recent Inventions as Applied to Modern Steamships, W. C. Wallace. (Friday) Service Test on Steamship Harvard, C. H. Peabody; U S Scout Cruiser Chester Fitted with Parsons Turbines, C. P. Wetherbee; The Steam Turbine, J. W. Powell; Shipbuilding on the Great Lakes, Robert Curr; The Steamer Commonwealth, J. H. Gardner, W. T. Berry; Fire Boats, C. C. West; Sea Going Suction Dredges, T. M. Cornbrooks; British International Trophy Race of 1908, W. P. Stephens; Transportation of Submarines, W. J. Baxter.

SOUTHERN AND SOUTH WESTERN RAILWAY CLUB

November 19, Atlanta, Ga. Secy., A. J. Merrill, 218 Prudential Bldg.

ST. LOUIS RAILWAY CLUB

December 11, monthly meeting. Secy., B. W. Frauenthal.

TECHNICAL SOCIETY OF BROOKLYN

November 20, December 4, 18, semi-monthly meetings, Arion Hall, Arion Place, Brooklyn, N. Y. Secy., Karl Koelble.

TECHNICAL SOCIETY OF THE PACIFIC COAST

December 4, bi-monthly meeting at San Francisco, Cal. Secy., Otto von Geldern, 1978 Broadway.

TECHNOLOGY CLUB OF SYRACUSE

December 8, monthly meeting, 502 The Bastable Blk., 8 p.m. Secy., Geo. D. Babcock. Paper: Coast Defence and Effect of High Explosives on Armor Plate, C. S. Jamieson.

WASHINGTON SOCIETY OF ENGINEERS

November 24, Hubbard Memorial Hall, 8 p.m. Secy., John F. Hoyt, 1330 F. St. N. W. Paper: Utility of the Water Resources of the Hawaiian Islands, F. H. Newell.

WESTERN ELECTRIC CLUB

December 8, 29 W. 39th St., New York, 8 p.m. Secy., J. W. Dietz, 463 West St.

WESTERN RAILWAY CLUB

December 15, monthly meeting, Auditorium Hotel, Chicago, Ill., 8 p.m. Secy., Jos. W. Taylor, 390 Old Colony Bldg.

WESTERN SOCIETY OF ENGINEERS

December 2, December 11, electrical section, December 16, extra meeting; 1737 Monadnock Blk., Chicago, Ill. January 5, 1909, annual meeting. Secy., J. H. Warder.

OFFICERS AND COUNCIL

1908

PRESIDENT

M. L. Holman ..... St. Louis, Mo.

VICE-PRESIDENTS

Alex Dow ..... Detroit, Mich.
P. W. Gates ..... Chicago, Ill.
J. W. Lieb, Jr ..... New York

Terms expire at Annual Meeting of 1908

L. P. Breckenridge ..... Urbana, Ill.
Fred J. Miller ..... Center Bridge, Pa.
Arthur West ..... E. Pittsburg, Pa.

Terms expire at Annual Meeting of 1909

PAST PRESIDENTS

James M. Dodge ..... Philadelphia, Pa.
Ambrose Swasey ..... Cleveland, O.
John R. Freeman ..... Providence, R. I.
Frederick W. Taylor ..... Philadelphia, Pa.
F. R. Hutton ..... New York

Members of the Council for 1908

MANAGERS

Walter Laidlaw ..... Cincinnati, O.
Frank G. Tallman ..... Wilmington, Del.
Frederick M. Prescott ..... Milwaukee, Wis.

Terms expire at Annual Meeting of 1908

A. J. Caldwell ..... Newburg, N. Y.
G. M. Basford ..... New York
A. L. Riker ..... Bridgeport, Conn.

Terms expire at Annual Meeting of 1909

Wm. L. Abbott ..... Batavia, Ill.
Alex. C. Humphreys ..... New York
Henry G. Stott ..... New Rochelle, N. Y.

Terms expire at Annual Meeting of 1910

TREASURER

William H. Wiley ..... New York

HONORARY SECRETARY

F. R. Hutton ..... New York

CHAIRMAN OF FINANCE COMMITTEE

A. W. Burchard ..... Schenectady, N. Y.

SECRETARY

Calvin W. Rice ..... 29 W. 39th Street, New York, N. Y.

STANDING COMMITTEES

1908

FINANCE

ANSON W. BURCHARD (1), *Chairman*
ARTHUR M. WAITT (2)
EDWARD F. SCHNUCK (3)
J. WALDO SMITH (4)
A. C. DINKEY (5)

MEETINGS

CHAS. WHITING BAKER (1), *Chairman*
W. E. HALL (2)
WM. H. BRYAN (3)
L. R. POMEROY (4)
CHARLES E. LUCKE (5)

MEMBERSHIP

JESSE M. SMITH (1), *Chairman*
HENRY D. HIBBARD (2)
CHARLES R. RICHARDS (3)
FRANCIS H. STILLMAN (4)
GEORGE J. FORAN (5)

PUBLICATION

FRED J. MILLER (1)
WALTER B. SNOW (2)
D. S. JACOBUS (3)
H. F. J. PORTER (4)
H. W. SPANGLER (5)

LIBRARY

A. W. HOWE (1)
H. H. SUPLEE (2), *Chairman*
AMBROSE SWASEY (3)
LEONARD WALDO (4)
G. M. BASFORD (5)

EXECUTIVE

M. L. HOLMAN
F. W. TAYLOR
J. W. LIEB, JR.
FRED J. MILLER
CALVIN W. RICE

NOTE—Numbers in parentheses indicate length of term in years that member is yet to serve.

SPECIAL COMMITTEES

1908

On a Standard Tonnage Basis for Refrigeration

D. S. JACOBUS — G. T. VOORHEES
A. P. TRAUTWEIN — PHILIP DE C. BALL
E. F. MILLER

On Society History

JOHN E. SWEET — H. H. SUPLEE
CHARLES WALLACE HUNT

On Affiliated Societies

F. R. HUTTON, *Chairman* — F. W. TAYLOR
R. H. FERNALD — H. H. SUPLEE
ALEX. C. HUMPHREYS

On By Laws for Research Committee

CHAS. WALLACE HUNT, *Chairman* — F. R. HUTTON
G. M. BASFORD — D. S. JACOBUS
JESSE M. SMITH

On Increase of Membership

WILLIAM C. DICKERMAN, *Chairman* — CHARLES E. LUCKE
GEORGE J. FORAN — FREDERICK M. WHYTE

On Conservation of Natural Resources

GEO. F. SWAIN, *Chairman* — L. D. BURLINGAME
M. L. HOLMAN — CALVIN W. RICE
CHARLES WHITING BAKER

On International Standard for Pipe Threads

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